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DEDICATION

TO

MY FAMILY, MY WIFE AND MY KIDS

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HYDRAULIC MACHINES OPERATION SIMULATION AND CONTROL

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ABSTRACT

This Thesis presents theoretical and experimental investigation on process control and simulation of hydraulic machines processes, where centrifugal compressors are the major consumers of the energy in the industrial plants. They used to provide refrigeration or to transport gases including hydrocarbon gases for natural gas liquefaction plants (NGL) and liquefied petroleum gas (LPG) plants. It considered as the most common means of gas compression used nowadays. The main research effort focused on operation stability and transient operation came across particularly in centrifugal compressor operation known as *Surge* phenomenon, theoretical modifications have been done to predict *Surge* based on transient response analysis. A process monitoring and control (PMC) simulator, represented by a hydraulic circuit chosen as a sample process. Utilizing data acquisition system (DAS) and PC computer with relevant software packages designed for this research and similar applications.

The behavior of the centrifugal compressor stage analyzed, and also the fluid behavior during compression process that leads to the *surge* onset investigated. The theoretical modeling and numerical calculations for steady state condition done. Operation monitoring and logging of the compressed air system flowing to the hydraulic bench presented. monitoring the unsteady operation of the compressed air through the test rig and logging of *surge* onset during both start up and throttling procedures. Plot the experimental data on charts to compare this data with theoretical data for the same process (test rig) done. Finally the benefit of this research actually achieved by employing the results to construct a suggested process control system to overcome *surge* onset.

The obtained results show a reasonable agreement between the theoretical model based on Greitzer's and Hansen's models and the simulator experimental results, for prediction and onset of *surge*. The model will be efficient and economic means of process monitor for conditions similar to the real industrial plants.

ABBREVIATIONS

SYMBOL	DEFINITION
A/D	: Analog to Digital converter.
API	: American Petroleum Institute.
AS	: Air Supply
BASIC	: Beginner's All Purpose Symbolic Instruction Code.
DAS	: Data Acquisition Systems.
D/A	: Digital to Analog converter.
DC	: Direct Current.
DCS	: Distributed Control System.
dPT	: Differential. Pressure Transmitter.
EM	: Electric Motor.
ES	: Electric Supply.
ESD	: Emergency Shut Down.
FCV	: Flow Control Valve.
FT	: Flow Transmitter.
H	: Hydraulic.
K	: Orifice Coefficient.
K1	: Air Compressor.
KB	: Key Board.
MCR	: Multi Component Refrigerants.
MIMO	: Multi I/P Multi O/P.
MISO	: Multi I/P Single O/P.
MUX	: Multiplexer.
NBS	: National Bureau of Standards.
NPRA	: National Petroleum Refiners Association.
O.P	: Orifice Plate.
PCV	: Pressure Control Valve.
PD	: Proportional plus Derivative Control Mode
PI	: Proportional Plus Integral control Mode
PID	: Proportional Plus Integral plus Derivative control Mode

PMC	: Process Monitoring and Control.
PT	: pressure Transmitter.
PV	: Process Variable.
RM	: Rotameter.
RAND	: Random.
RTD	: Resistance Temp. Detector.
SISO	: Single Input Single Output.
TC	: Thermocouple.
TI	: Temperature Indicator.
XIC	: Compressor Performance Controller.

NOMENCLATURE

SYMBOL	DEFINITION	ENGINEERING UNIT
a	: Speed of sound	(mps)
A	: Flow through area	(m^2)
bps	: Baud Rate per sec.(Bits per sec.)	
B	: Greitzer's compressor stability parameter	(non-dimensional)
C	: Compressor pressure rise	(non-dimensional)
°C	: Degree Celsius.	(non-dimensional)
e	: Error signal	(non-dimensional)
E	: Voltage.	(Volt)
F	: Throttle pressure drop	(non-dimensional)
G	: Geometric parameter	(non-dimensional)
h	: Differential pressure across orifice.	(cm H_2O)
I	: Current.	(mA)
KPa	: Kilo Pascal = Kilo Newton / Sq. meter	
L	: Duct length	(m)
M	: Molecular weight.	(mol.)
m	: Mass flow rate.	(m^3/sec)
mA	: Milli-Ampere.	
mV	: Milli-Volt.	
N	: Compressor Speed	(r. p. m.)
P	: Pressure	(KPa)
U_o	: Absolute tip blade speed	(r. p. m.)
U_i	: Absolute hub speed	(r. p. m.)
r	: Impeller Radius	(m)
R	: Ideal Gas Constants.	(N-m/Kg mol.°K)
S_n	: Variable Signal as % of Span (S_1, S_2, S_3 & S_4).	(non-dimensional)
(S)	: Hansen's valve parameter { $S = (A_c / A_t)^2$}	(non-dimensional)
SP	: Set Point.	(non-dimensional)
V_p	: Exit plenum volume.	(m^3)
C_t	: Tangential velocity	(mps)
C_r	: Radial Velocity	(mps)

W	: Gas Velocity Relative to Blade	(mps)
α	: Absolute angle (Exit of gas)	(degree)
ρ	: Density.	(Kg/m^3)
ω	: Helmholtz resonator frequency	(non-dimensional)
ω	: Uncertainty	(non-dimensional)
ΔP	: Plenum pressure rise	(non-dimensional)
τ	: Compressor flow time constant	(non-dimensional)

Subscripts :

p = Plenum   c = Compressor   t = Throttle   ss = Steady state condition

Computer-based process monitoring and control (PMC) becomes essential nowadays, process control of hydraulic machines in the chemical and petroleum industries is rapidly changing towards computer-based control systems. pneumatic control systems that have been traditionally used in chemical processing plants including rotating machinery such as pumps and compressors, are giving way to the electronic and computer control systems, to gain accuracy or precision and productivity, efficiency and savings.

PMC gives continuous, moment attentions to the process control applications to the degree impossible to be achieved by human operators, PMC can compute, remember, reason and forecast beyond the data programmed into them. PMC has a number of advantages over conventional old panel board control systems:

1) Continuous monitor conditions and makes necessary changes to assure that the process operates in the most efficient effective manner.
2) Rapid speed of response based on measuring and anticipating changes in operating conditions.
3) Easy maintenance diagnostics which will cut the shut-down periods.
4) Ensuring the stability of the automated process.
5) Optimizing the performance of the automated process.
6) Avoid human faults allowing to use expert systems.
7) Suppressing the influence of the external effects.

Turbo Compressors are major users of energy in industrial plants, They can be used as noticed in the following manner:

Firstly to produce compressed air for: plant air supply, separation plants, catalytic cracking processes, ammonia plants nitric acid plants, gas turbines (combustion & cooling air).Secondly to provide refrigeration {hydrocarbon gases}.Third to transport gases {such as hydrocarbon gases} in NGL & LPG, refinery plants and gas injection plants. turbo compressors create a head by imparting momentum from an impeller to the gas, such as centrifugal compressors. [Elliot, 1977]. Nowadays the modern trend of hydraulic machines processes dictates, for economic reasons is to replace the Positive displacement type compressors with centrifugal compressors, these dynamic compressors are essentially as a constant pressure, variable volume machines (as contrasted to a positive displacement compressors being a constant - volume, variable- pressure machine) and must be regulated in almost all applications.

Most hydrocarbons' process uses the centrifugal compressors that become one of the most common means of gas compression used today's. Its main advantages are mechanical simplicity compared to reciprocating and some other forms of rotary compressors, this generally reflects lower initial costs and lower maintenance costs. Each type of compressors exhibits application and control problems that are unique, centrifugal compressor *Surge* is unique and is a problem in most applications.

One of the most damaging aspects of compressors control is the (*Surge* protection), the challenge comes from the fact that is impossible to determine accurately the approach of *surge*.

Surge consists of large amplitude oscillations of the flow through the entire compressor that is also produce large oscillations in a compressor delivery pressure. Once a compressor begins to *Surge* it will continue until corrective action is taken, so automatic protection is mandatory. Preventing *Surge* from developing at all requires a control system that skirts the unstable operating area completely.

1.1 Research Motivation:

The reasons for automation of the Hydraulic machine's process operation is to satisfy several requirements imposed by its designers and the general technical, economical, and social conditions in the presence of ever-changing external disturbances [Stepf, 1984]. Among such requirements are: safety operation, production specifications, environmental regulations and operational constraints. Centrifugal compressor must operate in the safe limits away of *surge* limit and overload limit, control systems are needed to satisfy all these constraints.

Digital techniques are preferred to be used in industrial processes than analog techniques [Beach, 1983] because of; greater accuracy, greater dynamic range, greater stability, convenience, automation, and design simplicity computer-based process monitoring and control (PMC), brings consistency and reliability which translate into higher production yields-to chemical and petrochemical plants [Gauther, 1982].

1.2 Problem Definition:

The objective of this thesis is to study the Instability operation of turbomachinery in industrial plants, especially centrifugal compressors that are the major consumer of energy in the industrial plants. these centrifugal compressors are limited by a *surge* region where unstable operation is encountered at low flow rates, for these reason flow levels must be maintained at high enough levels to keep the compressor out of *surge* region where process demand is low.

Surge is an unstable condition within turbo compressor's blades. When the hydraulic energy losses for a given turbo compressors are equal to energy gained, a large oscillation of the total mass flow rate through compressor occur, and flow reverse through impellers produces a violent torque reversal which may have a damage in the rotating elements. Also it produces a reverse thrust load that thrust collar cannot overcome and damaged. This phenomenon will be analyzed, discussed, for multi stage centrifugal compressor used to compress hydrocarbon gas. The mass flow rate required to achieve a safe operation away of *surge* and away of stone wall on the performance curve of compressor will be determined. Transient response of a certain compressor depends on non-dimensional parameters investigated. System modeling during unsteady operation is done by means of numerical simulation using non-linear lumped parameters for on line prediction of *surge and rotating stall* utilizing expert system. Computer-based process monitoring and control (PMC) are essential in such cases for the purpose of minimize energy costs and protecting compressors from *surge*.

Due to the difficulties of applying the PMC in this research on a real large centrifugal compressor process, a similar custom process control system was

assembled represented by a hydraulic circuit as a sample process. Data acquisition system (DAS) and PC computer (AT) with relevant software were used for this research. The above tools constitutes the lab hydraulic circuit control and simulation (simulator). The previous constructed PMC provides an efficient and economic mean to simulate real large industrial process.

1.3 Previous Work:

The computer-based-process control systems have the potential of becoming the most significant technological development in hydraulic machine's industry [Schmitt, 1984]. The wide spread application of digital computers for hydraulic machine's process control is the culmination of a 15-years struggle for acceptance, beginning in 1958 when the first industrial control computer was introduced.

The role of the computer has progressed from most elementary function of collecting and recording data, manually entered by the process operator to complete plant-wide automatic optional control. This role is a part of the revolutionary trend towards more continuous processing and less dependence on manual control Parametric information is a prerequisite for controlling a process. As process becomes more sophisticated, the volumes of information represent a significant time lag for operational analysis and decision making.

A computer based hydraulic machine's process was successfully implemented in 1982 [Farouqi, 1982]. The project's feasibility was studied to find out how and where the computer would benefit the hydraulic machine's process through improved control and optimization. Through process experiments, extensive lab analysis and

computer simulations, that computer control yields the greatest profits in several ways such as:

1- Increases the product recovery through better control of the units.

2- Decreases the internal recycle streams and energy cost.

3- Increase the availability for operating personal of more complete information on the production situation and the functioning of the process.

The analysis of *surge and rotating stall* (which occur directly before *surge*) was first observed by the group developing centrifugal compressor for whittle turbojet in (1938) [Chesir L. J., 1954]. A comprehensive list of publications on rotating stall up to 1967 has been assembled by [Fabri, J., 1967]. A series of ASME papers extended from 1954 up to 1972 cover much of the significant development in this field [Emmons H. W. 1955, 59], [Lura, 1954], [Stenning, 1958,1980] and [Takata H., 1972]. This analysis in which had been done is to describe the theory of the onset of *rotating stall and surge*, but does not yield a criterion for the onset of rotating stall or *surge* that could be applicable to single stage compressors and multistage compressors.

[Metcalf, 1972] represent how to calculate and eliminate *surge* using Daniel flow computation techniques and using relays control scheme on 1971. [Dunham, 1965] found that rotating stall followed by *surge* occurs at the peak of the performance curve of centrifugal compressors that separates stable operating area than unstable operating area. [Whitfield, 1976, 1977] agrees that difficulties of predicting the flow through centrifugal compressor is often highly separated and fully three dimensional.

He also states that up to this time, the computing techniques have not provided the ability to predict this three - dimensional separated flow.

Whitefield and Coppage and Balje; defined the losses through a stagnation enthalpy loss coefficient nondimensionalized by the tip speed of the impeller. Also Jansen and Redgers uses the same technique in addition to using the stagnation enthalpy loss and stagnation pressure loss coefficient and efficiency decrement in order to calculate the flow losses. [Brown L. E., 1972] showed that as mach number increased several losses increased. [Takata H. And Nagano, 1971] had undertaken a numerical solution of the non linear unsteady equation of motion for rotating stall that has been reproducing most of the observed phenomena. He indicated that the number of stall cells in a complete stage governed by the interference effects between blade rows but they have not been able to explain why different numbers of stall cells appear at different operating conditions in an isolated rotor.

[Stenning, 1980] made a good analysis for both stall and *surge*. he developed the stall onset criterion and the velocity of propagation of stall cells in a cascade. this accomplished by dividing a multistage compressor to a number of stream tubes (5 to 9) and calculating rotating stall for each tube and blade raw. He found that *surge* occurs when the streamtube close to the hub is unstable for rotating stall. Consequently a number of performance prediction procedures with a general assumption of one-dimensional adiabatic flow have been published [ASME Rationalization]. [Balje O. E. (1952) and Coppage J. E. (1956) and Whitfield (1976)].

The first technique was done by [Stening, 1980] using linear differential equations, linearizing the compressor characteristic in the vicinity of the steady state

operating point, using simple lumped parameter mode [Stening, 1980], assumed a pressure drop from the plenum to the atmosphere as a function of the mass flow through valve. This represents a small disturbance away from the steady state, then he linearized these equations ~~were~~ lineariz~~ed~~ation lead~~ing~~s to his characteristic equation. This technique of [Stening] is a good approach to model the *Surge* but is not accurate enough for simulation using expert system for predicting *surge* and allowing anti *surge* control to over come *surge*.

The second modeling technique was done by [O'Brine W. F., 1990] using partial differential equations that are far accurate but at the some time far long and slow to be implemented in industry by means of expert system. [O'Brien W. F., 1990] used distributed parameters which needs parallel computers to simulate the unsteady condition, this technique formulates a stage - by stage dynamic system model which included details of internal features of a compression system such as effects of off - design stage mismatching, interstage bleeds, and variable geometry could be studied. This means the difficulties of implementing this technique in *expert system* applications for similar cases.

The third technique was done by [Grietzer, 1975] allowing the use of non - linear lumped parameter to develop a non - linear model for predicting the transient response of a compression system. He developed a non dimensional parameters on which this response depends. For value above the critical, the system will exhibit the large amplitude oscillatory behavior characteristic of *surge*, while for values below the critical, it moves toward operation in rotating stall at a substantially reduced flow rate and pressure ratio.

The overall dynamics of *surge* and rotating stall in axial flow compression systems have been studied theoretically and experimentally by [Grietzer, 1975]. A non-linear lumped parameter model was found to describe a variety of large amplitude oscillations in the real time domain. Dupuis and others in 1986 analyze the performance of low pressure compressor and high pressure compressor. He was able to predict operation fault by means of engine health monitoring (E.H.M) depending on the changes of turbine speed through nozzle adjustment and turbine efficiency loss. However, neither a complete picture of mechanisms at play, nor an applicable quantitative flow model describing the rate of flow break down appears to be available at this time to assist in the prediction of overall *surge* behavior [Hansen, 1981], presents study depending on Greitzer 's work to ~~Farther~~ explore the greitzer model and test its applicability to describe *surge* in a small single-stage centrifugal compressor system. The dynamic behavior prediction based on measured steady- state branch of centrifugal compressor characteristics was tested.

further

1.4 Plane of Research:

The research work (*surge* instability investigation) will be accomplished through two phases as following:

i) Theoretical modeling phase:

Grietzer, 1975 developed a Non - linear lumped parameter model to predict the transient response of a compression system subsequent to a perturbation from steady operating conditions. This non-linear lumped parameter was found to describe a variety of large amplitude oscillations in the real time domain. In this research a modified Greitzer and Hansen modeling are proposed to increase accuracy of the solution by solving the four non - linear differential equation ~~will be done~~ using fifth - order Rung - Kutta integration for real case.

ii) Experimental and simulation phase:

The experimental investigations take two paths;

1. First path will be the monitoring of the proposed test rig used in this lab. it composed of; compressed air supply, hydraulic circuit, with obstruction flow measuring devices and thermocouples (type K, NBS), and sensitive pressure transducers (variable reluctance type) the monitoring achieved by registering the sensors reading by the DAS and PC computer. This cover real time monitoring and logging of the system operation during steady state condition. This system will be the process monitoring & control system (PMC) which will utilize a software program written in Quick Basic (QB V-4).
2. The second path is the monitoring of the same system during unstable operation that yields to the *surge* onset, the evidence of the onset of the *surge* will be visually

and audibly observed on the glass tube of Rotameter (flow measuring device constructed on the test rig).

A series of computer simulations were performed using the theoretical models developed in phase one obtained according to the conditions existing in the experimental set up used in phase two. A through investigation for comparing both the simulation and actual results will be done to test the feasibility of the modified techniques.

2.1 *Surge* Phenomenon:

Surge is an unsuitable condition within the compressor blades. A centrifugal compressor head-flow curve typified by the 100% speed line (Fig. 2-1)[Elliot, 1977]. As gas flows through the compressor decreases from max. the compression ratio increases to a max. and falls off again with further decrease in flow. When the flow through the compressor falls below this max. pressure ratio point, the flow through the blades becomes unstable. This point called the *surge point* of the compressor. [Metcalf, 1972] For each compressor speed there is a characteristic head-flow curve with its characteristic *surge* point. The line through the *surge* points at various compressor speeds called the *surge Line* as shown in (Fig.-1).

This unstable flow condition in a centrifugal compressor is usually noisy and frequently violent enough to damage the compressor or associated piping, therefore, in most compressor applications where the flow may drop below the *surge* Line, it is important to provide a control system that will protect the compressor.

In most cases it is possible for the process to force the compressor into a *surge* condition very quickly. This means that the *surge* control system must design with maximum emphasis on speed of response. [Vandaveer, 1982, Metcalf, 1972]

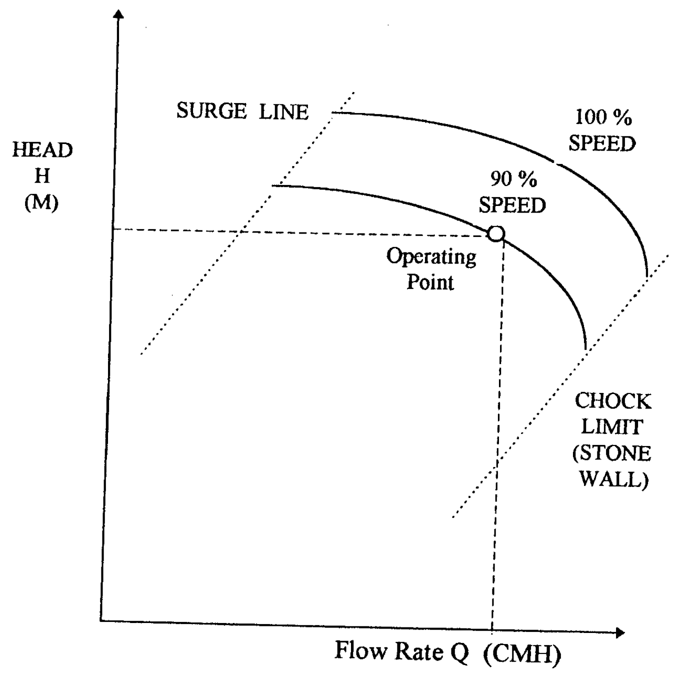


Fig.(2-1) Performance curve for Centrifugal Compressor.[Elliot, 1977]

2.1.1 *Surge* region:

Centrifugal compressors limited by *surge* region where unstable operation is encounter at low flow rates. In the (*surge* region) the head - flow characteristics of the centrifugal compressors actually is a reverse slope. As flow reduced, discharge pressure falls, causing flow and pressure to be further reduced.

When discharge pressure falls below that in the line, a momentary reversal flow occurs, and line pressure starts to fall. This creates a demand for more flow, causing flow to reverse again, the rapid pulsation will continue until the control system moves the compressor out of the *surge* region, or until the compressor damaged and falls (shut down).

Flow levels must be maintained at high enough levels to keep the compressor out of the *surge* region when the process demand is low. This flow may be provided by recirculating cooled discharge gas back to the suction through a bypass control valve that represent a waste of energy for a fixed speed driven compressor. A variable speed driven can extend the operating range of the compressor and should be used when wide ranges in operating conditions are expected.

If compressor speed is not (variable throttled) of the suction using inlet guide vanes or throttling valve normally used as an alternative means of control. The vanes reduce suction pressure, flow and discharge pressure and provide a pre-rotation of the gas. Generally *surge* occurs when the hydraulic losses equal to the gained energy, specifically, *surge* in a centrifugal compressor occurs when a compressor wheel is unable to overcome the resistance of the system to gas flowing through the

compressor. Resistance to gas flow is attributable to a combination of rotor and stator energy losses *[Patlovany, 1986]*.

Rotor losses include; Inducer incidence angle losses, diffusion blading losses, skin friction losses due to shear forces in the boundary layers, clearance losses and recirculating losses. stator losses include: skin friction losses in the diffuser, wake mixing losses caused by the (wake) of an impeller blade as it moves into radial vane's space. All of these losses rob energy from the gas flow. As the gas flow decreases, portions of the compressor will stall, and if the stall is sufficient strength, the compressor will *surge*. For a given compressor speed, the higher the fluid velocity through the machine, the higher the rotor and stator energy losses.

2.1.2 S*urge* prediction:

S*urge* prediction will help the operator to run the compressor at the lowest gas flow through the compressor, without fearing of *surge* event that could cause serious damage to the compressor, with minimum gas recirculation or air vented that will save the energy wasted. this potential energy savings can reach 18% per year. To operate the compressor as close as possible to its *surge* line, the compressor *surge* prediction, must conduct in a stage by stage analysis to take into account wheel designs, intercooling, and piping con*figuration.* to determine exactly which stage of the compressor will *surge* at the highest flow rate. The stage that *surge* s with the highest flow rate is the limiting factor on how the entire compressor can run close to its *surge* line.

surge line prediction on a stage by stage basis on a machine cannot done by simply running the machine at varying discharge pressures and suction temperatures and decreasing either speed (for turbine driven) or decreasing the suction flow rate (for motor driven), until a *surge* event is recorded. This can only create a *surge* map of the machine as a whole, and is liable to the accuracy because the onset of the *surge* in a single stage of a compressor is rarely noticeable audibly, and only becomes so when its instability causes the rest of the machine to *surge*.

To conduct the performance tests of a given compressor, data must be obtained for three sections to check the accuracy of the stage by stage *surge* line predictions that make up each section. Information required for an analysis program in the prediction of compressor performance includes physical dimensions of rotor and diffusers, dynamic pressure and case vibration data, flow and speed.

A solution of inlet and exit velocity triangles must be performed for calculation of head and discharge pressure. Head losses in the impeller (Rotor Losses) suggested to be calculated, and an impeller iteration on the velocity triangles to calculate real conditions versus ideal conditions.

Knowing the impeller discharge pressure, velocities through the diffuser must be computed and head losses then used to calculate a diffuser efficiency, and stage exit pressure. *surge* points then calculated by generating a velocity profile for flow through the impeller and *surge* said to occur when the velocities indicate separation of the flow from the wall (a stall). The *surge* point calculated in a dimensionless parameter, which can then relate back to any flow, pressure, speed, or to temperature conditions. Detailed performance (maps) and *surge* line predictions can be generated.

Data on the subject compressor collected by mounting dynamic pressure transducers on bleed taps of each stage, and by mounting accelerometers on a compressor case in both horizontal and vertical planes. Data of speed, flow and temperatures recorded manually during operation, Diffuser and rotor dimensions collected while disassembling the compressor for annual maintenance.

2.1.3 *Surge* detecting:

An actual *surge* event detected in three ways: [Vandaveer, 1982]

a. Detecting a sudden drop in discharge flow.

b. By detecting a sudden increase in inlet pressure of any compressor section, or

c. By detecting a sudden increase in inlet temperature for any of the compressor section.

Any of these three incidences indicates that *surge* is already occurring and that *surge* avoidance must be taken immediately. *surge avoidance* actions itself is simply ramping the vent or recycles valves open to increase gas flow and boosting compressor speed so as not to greatly affect the discharge pressure. If *surge* event recognized, the controller must rise the predicted *surge* flow by one percent.

2.2 Qualitative Analysis:

surge analysis starts with the inside picture of the centrifugal compression stage that can help to determine what can realistically be expected of centrifugal compressor and Analysis of competitive offerings. The discussion here concerned with

conventional compressor stage with radial inlet, closed impeller running at tip speed (U) of 200 to 300 (mps), feeding a vanless diffuser. Impeller and diffuser are the two key elements in producing characteristic shape of compressor performance curve. Poorly designed inlets, collector (return channels) can naturally affect performance but their influence on characteristic shape is usually small and will be ignored (*Fig.* 2.2) [Elliot, 1977].

Referring to *Fig.* (2-1) performance curve of centrifugal compressor there are: the basic slop of head *h* Vs flow *Q*, stone wall (chock limit) and *surge* limit at minimum Flow. The impeller tip velocity vectors could be drawn as in Fig. (2-3) W represents gas velocity relative to the blade, U_2 represents the absolute tip speed of the blade, the result of the two vectors is represented by C, which is the actual absolute velocity of gas (by vector addition, $U_2 + W = C$) It can be seen that the length of the vectors and the magnitude of the exit angle and are determined by the amount of back word lean in the blade, and by gas velocity relative to the blade, which by the tip speed of the blade U_2 is in turn dictated by tip- volume - flow rate for a given impeller. Having the magnitude and direction of the absolute velocity C breaking this vector into its radial and tangential components: C_r and C_t (Fig. 2-4A).

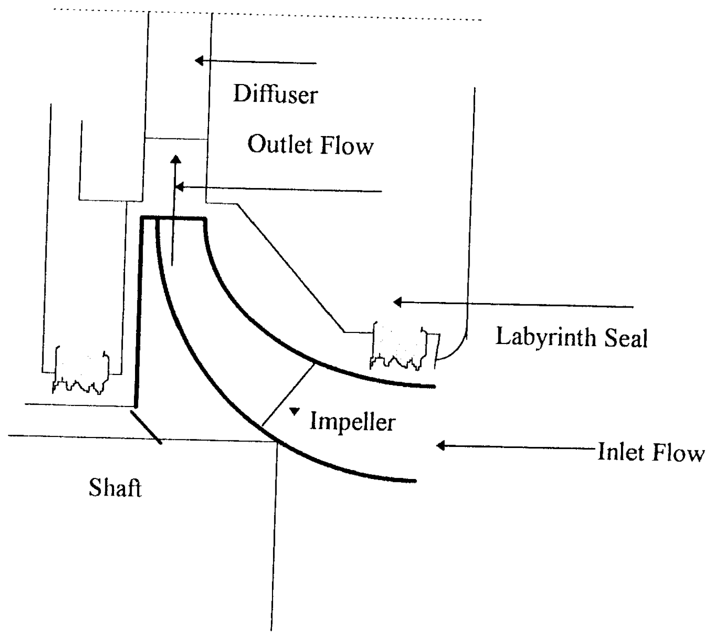


Fig. (2-2) Centrifugal Compressor Impeller[Elliot, 1977]

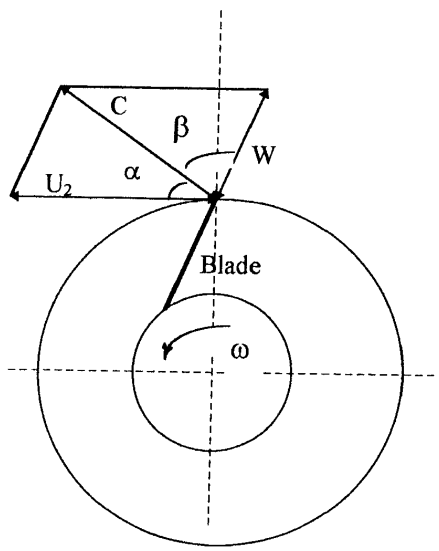


Fig. (2-3) Blade Velocity Triangle [Elliot, 1977]

C = Absolute Velocity .... MPS
W= Relative Velocity .... MPS
U = Blade Velocity .... MPS
α = Absolute Angle
β= Relative Angle

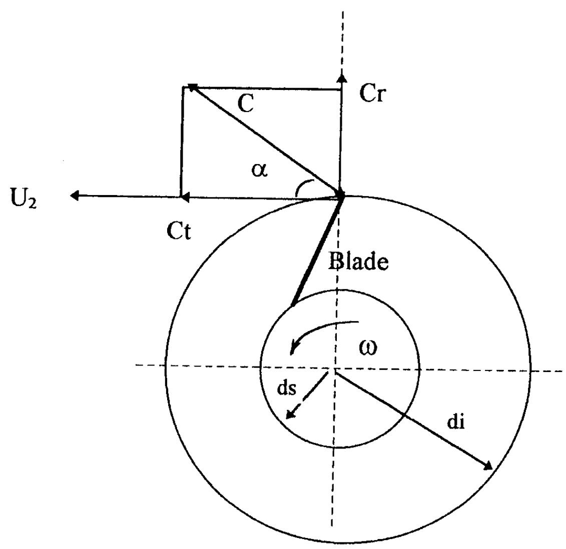


Fig. (2-4A) Blade Velocity distribution

C = Absolute Velocity .... MPS
Cr = Radial Velocity .... MPS
Ct = Tangential Velocity .... MPS
ds = Shaft diameter ....mm
di = Impeller Tip diameter ....mm
α = Absolute Angle

For an impeller running at constant rpm, As flow is decreased, W decreases, and angle (α) decreases also. This makes C_t increases which increases head out put, this head increase with decreasing flow. In the normal parallel-wall vaneless diffuser, this angle remains almost constant throughout the diffuser so the path taken by a particle of gas is a log spirals as in *Fig.* (2-4B). The reason that angle (α) remains constant in a parallel-wall diffuser is that both C_r and C_t vary inversely with radius, because radial flow area is proportional to radius and C_t .

It is evident from *Fig.* (2-4A) that the smaller the angle (α), the longer the flow path of given gas particles between the impeller tip and the diffuser outer diameter. When angle (α) becomes small enough and the diffuser flow path long enough the flow momentum at the walls dissipated by friction to the point where pressure gained by diffusion causes a reversal of flow and *surge* results. The angle (α) at which this occurs a vaneless diffuser has been found to be quite predictable for various diffuser-impeller diameter ratios. The angle (α) at which *surge* occurs can be lowered somewhat by reducing diffuser diameter, but at the cost of some velocity pressure recovery So when the C_r and (α) becomes too small, *surge* will occur. Vanned diffusers are recommended to increase the performance of centrifugal compressors, *Fig.* (2-4B), because the vanes force the air outward in a shorter path then unguided gas would be taken, but not so short a path as to cause too rapid deceleration with consequent stream separation and efficiency.

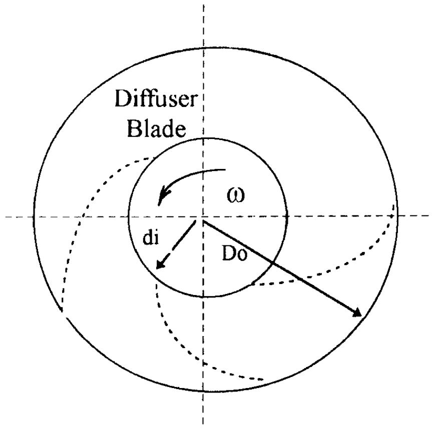


Fig. (2-4B) Blade Diffuser.

Do = Diffuser diameter ....mm
di = Impeller diameter ....mm

If flow is lower than design, the gas will hit the diffuser vanes with positive incidence that triggers *surge* at decreasing flow, inversely choking will be occur before (vanned diffuser) stone wall line. Back to the case of vanless diffuser when the C_r and (α) become too small, *surge* will occur, the solution is artificially increasing C_r and (α) by pulling (narrowing) the diffuser walls together until (α) reaches the proper value at design flow.

Shortly from previous analysis it is seen that as flow velocity C_r and angle (α) are reduced in the radial diffuser passage, the resultant velocity pressure drops off the extent where static pressure no longer held in balance and the flow suddenly reverses. Depending on the volume of the discharge pipe system and the size and speed of the compressor, the normal flow pattern quickly re-established and the sequence repeated until the condition ultimately corrected. It is often necessary to open the recycle valve (Compensate) supply enough amount of gas to suction side to avoid the pulsation and audible educable evidence of operating in the *surge* region which led to safe guard equipment.

Operating in the *surge* region for extended periods could result in mechanical failure due to *overheating* and *high stresses* depending on the energy level involved. (For fixed speed compressor motor driven) but for variable speed compressor (turbine driven), decreasing the speed of the unit respecting all turbine speed limitations, until problem identified corrected. choking also must be avoided by operating the compressor at $M<1$ (Mach number), where the W vector (magnitude of relative velocity divided by the acoustic velocity at the inlet to give relative inlet Mach no.) This ratio

must be held below 1.0 to avoid choking. This can be done by lowering the flow by closing the suction control valve partially.

2.3 Steady State Modeling:

The computing relays are black boxes that perform some mathematical manipulation of the signals. The blocks are software equivalent of the computing relays, with the advent of micro-processor based control system, starting in detail procedures as follow:

Linearization of the functions by considering the non linear function of two variables: [Carlos, 1984]

$$f[x(t), y(t)]$$

The TAYLOR series expansion around a point (~x, ~y) is given by

$$f[x(t),y(t)] = f(\bar{x},\bar{y}) + \frac{df}{dx}(\bar{x},\bar{y})[x(t)-\bar{x}] + \frac{df}{dy}(\bar{x},\bar{y})[y(t)-\bar{y}]$$

$$+ \frac{1}{2!}\frac{d^2f}{dx^2}(\bar{x},\bar{y})[x(t)-\bar{x}]^2 + \frac{1}{2!}\frac{d^2f}{dy^2}(\bar{x},\bar{y})[y(t)-\bar{y}]^2 + \ldots..$$

The linear approximation consists of dropping the second and higher order terms to obtain:

$$f[x(t),y(t)] = f(\bar{x},\bar{y}) + \frac{df}{dx}(\bar{x},\bar{y})[x(t)-\bar{x}] + \frac{df}{dy}(\bar{x},\bar{y})[y(t)-\bar{y}]$$

The error of the linear approximation is smaller for x, y in the neighborhood of ~x and ~y. In general; a function of n variables X1, X2...Xn is linearized by the formula:

$$f(X_1, X_2 \cdots X_n) = f(\overline{X}_1, \overline{X}_2 \cdots \overline{X}_n)$$
$$+ \frac{\overline{df}}{dx_2}(X_2 - \overline{X_2}) + \ldots + \frac{\overline{df}}{dx_n}(X_n - \overline{X_n})$$

Summarizing last equation to get:

$$\therefore f(X_1, X_2 \cdots X_n) = f(\overline{X}_1, \overline{X}_2 \cdots \overline{X}_n) + \sum_{k=1:n} \frac{\overline{df}}{dx_k}(X_k - \overline{X}_k)$$

Where: $\frac{\overline{df}}{dx_k}$ denotes the partial derivatives evaluated at (~X1, ~X2...~Xn).

Consider the density of Ideal Gas given by the formula:

$$\rho = \frac{MP}{RT} \quad \ldots.(3-1)$$

Where:

M: The molecular weight

M = 29 at ambient temp. And atm. Pressure.

R = The Ideal Gas constant for air .

= 8314 N-m /Kg mol. °K

P = Atm. Pressure 1 Bar

= 101300 N / m^2

The linear approximation of eqn. (2.3-1) according to previous procedure is given by:

$$\rho = \overline{\rho} + \frac{d\rho}{dT}(T - \overline{T}) + \frac{d\rho}{dp}(p - \overline{p}) \ldots(3-2)$$

The partial derivatives of the density function are:

$$\frac{d\rho}{dT} = \frac{Mp}{RT^2} = -\frac{\rho}{T} \quad \ldots\ldots\ldots\ldots\ldots\ldots (i)$$

$$\frac{d\rho}{dp} = \frac{M}{RT} = \frac{\rho}{p} \quad \ldots\ldots\ldots\ldots\ldots\ldots (ii)$$

This equation will be utilized for steady state experimental monitoring in next chapter substituting the equation with steady state condition operation data to get a useful proposed signal control equation.

2.4 *Surge* Theoretical Modeling (Qualitative Analysis):

Flow instabilities can be if two types: rotating stall and *surge*, the first describes bladding oscillating stresses while the second may have a disastrous effect on the whole system of which the compressor is a component [*Stenning, 1980*]. On *Fig.* (2-1) performance curve of centrifugal compressor for different speeds a Stall line separates stable operation from unstable operation. At unstable operation area to the left of stall line, large oscillations of the mass flow rate (*surge*) may occur. Self induced circumferential flow distortions may rotate around annulus (rotating stall). combination of both phenomena may appear *Fig.* (2-5a, b). This will produce a *rapid temperature rise* of the gas within the compressor in extreme cases causing blades to *melt* [*Stenning, 1980*] within a few seconds or minuet of the occurrence of instability. Although stall line represents a limit to the safe operation of the compressor, rotating stall can induce large vibratory stresses in the bladding of compressor. Theoretical technique of *surge* modeling during unsteady operations or transient response of compression system has done by three ways:

2.4.1 First modeling technique

It is developed by [Stenning, 1980] using linear differential equations, by linearizing the compressor characteristic in the vicinity of the steady state operating point, Stening has employed simple lumped parameter model for his technique *Fig.* (2-6). Stenning assumed a pressure drop from the plenum to the atmosphere as a function of the mass flow through valve in Fig(2-6). That represents small disturbances away from the steady state, then he linearized these equations that leads to his characteristic equation:

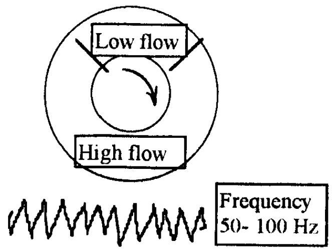


Fig. (2-5a) Rotating Stall onset

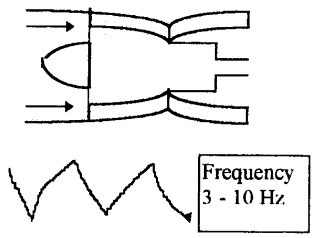


Fig. (2-5b) SURGE onset

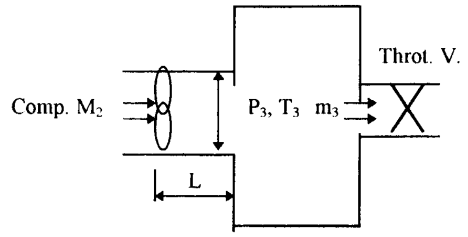


Fig. (2-6) Stenning Model

1- Pressure drop from the plenum to atmosphere is a function of exit flow rate:

$$F(m_3) = P_3 - P_{01}$$

2- Small disturbances away of steady state condition is:

$$P_3 - P_{01} = F' + (dF / dm_3)$$

3. Linearized equations for *surge* therefore are:

$$P_2 - P_3 = m_2 (L / A g_c)$$

$$m_2 - m_3 = P_3 (V / KRT_3)$$

$$P_3 = F' m_3$$

$$P_2 = C' m_2$$

$$\frac{LF'V}{A g_c k R T_3}\frac{d^2Z}{dt^2} + \left(\frac{L}{A g_c} - \frac{C'F'V}{KRT_3}\right)\frac{dZ}{dt} + (F' - C')Z = 0 \quad \text{..........(2-4-1)}$$

This equation is a second order equation, which has a constant coefficient if disturbances are small. Where:- ω = radian frequency

A = Flow area

$C = P_2 - P_{01}$

$F = P_3 - P_{01}$

g_c = coefficient. in Newton's law

K = Polytropic Exponent

R = gas constant.

T = outlet temp.

L = duct length

V = plenum volume

Z = variable

The coefficient of $d^2 Z / dt^2$ is always positive because, for any real value, F' is always positive. Hence instabilities occur if the coefficient of dZ / dt becomes (negative damping) or the coefficient Z becomes negative (Negative spring rate), if the coefficient dZ/dt becomes negative, the system is dynamically unstable and will undergo oscillations of increasing amplitude (*surge*). If the coefficient Z is negative, the system is statically unstable. If the valve is gradually reduced the steady state operating point moves to the left in the compressor characteristic (*Fig.* 2-7). $F' > C'$ and then coefficient Z always positive and C' is initially negative as soon as the peak of the compressor characteristic is passed.

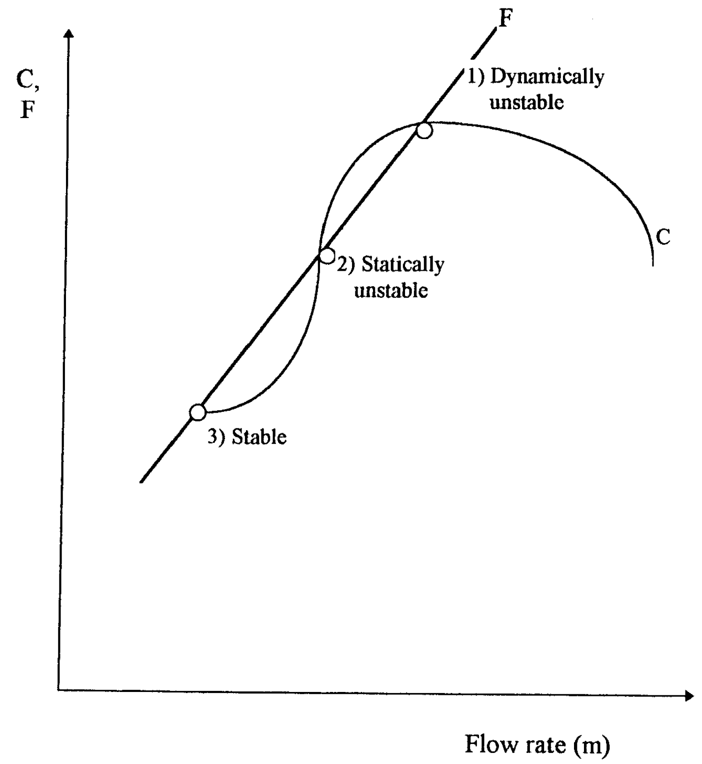


Fig (2-7) STENING PERFORMANCE CURVE INSTABILITY

$C' = \dfrac{dc}{dm}$ becomes positive

The system damping becomes Zero when :

$$\frac{C'F'V}{KgRT_3} = \frac{L}{Ag_c}$$

Or $C' = \dfrac{LKRT_3}{g_c AF'}$ ....(1 - 2) ....(2-4-2)

The *surge* oscillations then begin near the instability region, one or more stall cells travel around the compressor annulus in the direction of rotation of the compressor. Within each stall cell, the blades are so severely stalled that there is virtually no flow through the blade row. A stall cell may cover only a few blades, or may occupy a portion of the annulus, it may appear at the rotor tip of the bladding or may extent over the whole blade length.(*Fig.* 2-5) explains the illustration of rotating stall, consider a raw of compressor blades operating at high angle of attack, (*Fig.* 2-5a), if a flow disturbance produces a local increase in angle of attack on blade *B* (*Fig.* . 2-5b) then sever flow separation may occur. the resulting blockage of the channel will divert flow away from blade *B*, increasing the angle of attack or blade *A* and reducing the angle of attack on blade *C*, the stall will therefore propagate from right to left, if conditions are suitable, may build up into a fully developed Stall cells propagating along the cascade with velocity *Vp*.

[*Stenning, 1980*] indicates that high speed dynamic measurements of *surge* initiations in high pressure ratio compressors have indicated that *surge* may be triggered very rapidly by rotating stall , and that the slight dip in the compressor characteristic produced by the rotating stall may be sufficient to induce *surge*.

In low pressure ratio compressors, the stall line may be indicative of rotating stall alone before *surge* occurs. The *surge* oscillations may build up into large amplitude limit cycles as *Fig.* (2-7) the radian frequency of this oscillation is given by equation for a duct [*Stenning*, *1980*].

$$\omega = \sqrt{A g K R T / L V} \quad \text{.......(2-4-3)}$$

In some cases the system may find a new steady state operating condition as in *Fig.* (2-7) showing the three intersections P (1) of *surge* initiations which is dynamically unstable, P (2) is statically unstable and point (3) is stable, depending on the way in which *surge* is entered and the parameters of the system. It may be possible for the compressor either to *surge* continuously or after a few cycles to converge to point (3) on Performance curve. In this event severe compressor damage may be occur due to excessive temperature or to rotating stall if point (3) is in the region of rotating stall . that in which for compressor loss in efficiency due to flow decrease , causes a rise in temperature rise ratio (ΔT / T) at a given pressure ratio [Cohen, H. 1972] and[Dupuis,R. J. 1986]. This technique of Stenning. is a good approach to model the *surge* but is not *accurate* enough for simulation using expert system for predicting *surge* and allowing anti *surge* control to over come *surge*.

2.4.2 Second modeling technique:

This was done by [O'Brien W. F., 1991] using partial differential equations which is far accurate but at the same time far long and slow to be implemented in industry by means of expert system. O'Brien used distributed parameters that need parallel computers to simulate the unsteady condition. This technique formulates a stage-by stage dynamic system model which includes details of internal features of a compression system, such as effects of off - design stage mismatching , interstage bleeds , and variable geometry. He also used Mac Cromack predictor / corrector algorithm to solve his system equations of molecules, he advised the computational process required to complete one time step of the modified stage-by stage compressor simulation using parallel computer. This simulation has been run on many conventional serial computers, execution times were typically 100 - 500 times the time being simulated. this means the difficulties of implementing this technique in expert system applications for similar cases and long execution time of such technique utilizing the available personal computer in the labs.

2.4.3 Third modeling technique :

This was done by [Greitzer, 1975] (*Fig.* 2-8) allowing to use non-linear lumped parameter he developed a non-linear model to predict the transient response of a compression system subsequent to a perturbation from steady operating conditions. he developed a non dimensional parameters on which this response depends, for value above the critical. The system will exhibit the large amplitude oscillatory behavior characteristic of *surge*. For values below the critical, it will moves toward operation in rotating stall, at a substantially reduced flow rate and high pressure ratio.

Throttling the flow through an turbo compressor from design point to the stall limit, the steady flow pattern that exists becomes unstable. Instability is one of two forms either *surge* or rotating stall where *surge* is a large amplitude oscillation of the total annulus averaged flow through the compressor. The frequencies of *surge* oscillations are typically over an order of magnitude less than those associated which the passage of the rotating stall cells, (*Fig.* 2-5). In fact, during a *surge* cycle, the compressor may pass in and out of rotating stall as the mass flow changes with time.

On the other hand, if *surge* occurs, The transient consequences such as large inlet over pressures, can also be severe, however, the circumstances may well be more favorable for returning to unstalled operation by opening either the throttle or internal bleeds, since the compressor can be operating in an unstalled condition over part of each *surge* cycle. For this reason, one of the important problems associated with compressor stability is the determination of which of these two types of behavior will occur with a compression system.

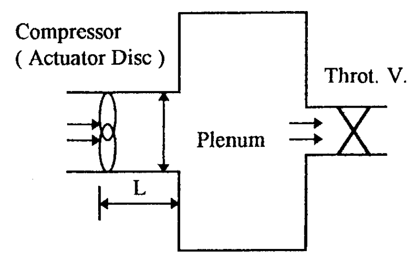


Fig (2-8)GREITZER Model

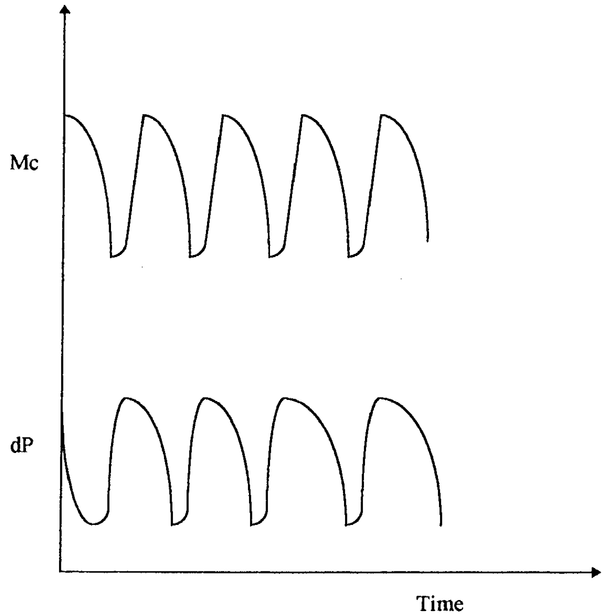


Fig (2-9)GREITZER's Transient
Compression behavior @ B = 1.58

2.4.3.1 Disadvantages of the third modeling approach:

The previous study indicates that neither a complete picture of mechanisms, nor an application quantitative flow model describing the rate of flow breakdown appears to be available at this time to assist in the prediction of overall *surge* behavior. [Hansen, 1981]. Nevertheless, for an engineering application in the system dynamics analysis a lumped parameter model, such as that of Greitzer used by [Hansen 1981]. Hansen presents study depending on [*Greitzer, 1975*] study to farther explore the *Greitzer* model and test its applicability to describe *surge* in a small single-stage centrifugal compressor system, to test the dynamic behavior prediction based on measured steady-state branches of centrifugal compressor characteristics. Hansen model compression system is open to the ambient at compressor inlet and down stream of the throttle valve (*Fig.* 10, 11). He also used (Hot-wire probes) positioned to the core of the flow at inlet and exit flanges, (Hot-wire extensively used for measurements of transient flows specially measurements of turbulent fluctuations) [Holman, 1984]. It is very useful for turbulence measurements because it can respond to very rapid changes in flow velocity, he also used strain gauges type pressure transducers for static and stagnation pressure measurements. The sample frequency of the DAS used by Hansen was 250 Hz. Hansen deduced that it is expected that the frequency of finite amplitude oscillations in a non-linear system is less than the natural frequency of the system considering it was (ω / 2π).

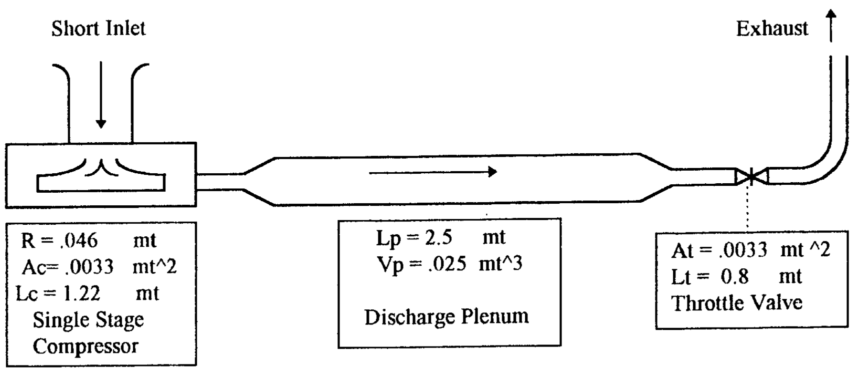


Fig (11) HANSEN MODEL
Single stage centrifugal compressor of turbocharger with vanned radial diffuser driven by turbine.

2-5 Hansen Modified Theoretical Modeling:

The schematic of the compression system model used in this analysis is shown in *Fig.* (2-9) far similar to system used by Stenning, The compressor and its ducting are replaced by an actuator disk to account for the pressure rise. This due to the Compressor and a length of constant one pipe for the dynamics of the fluid in the compressor duct.

Similarly the throttle also replaced by this combination of actuator disk, across which the pressure drops, in addition to a constant area duct. Greitzer assumed some approximations plus previous assumptions of actuator disks, these approximation are:-

1 - Oscillations associated with compressor *surge* has low frequency. (3 -10 Hz).

2 - The flow in the ducts can be *incompressible.*

3 - The flow density is equal to ambient values.

4 - The fluid axial velocity is constant (steady).

5 - Compressor responds quasi - steadily to changes in mass flow.

6 - Process in the plenum is Polytropic.

7 - The static pressure will be uniform through plenum at any instant time.

The present analysis based on a modified Greitzer and Hansen analysis. The analysis done using the non linear system behavior, using Helmholtz resonator type of compression system model introduced by [Emmons, 1959]. The analysis shows that for a given compression system i-e specified compressor characteristics, there is an important non-dimensional parameter on which the system response, responds , these parameters are denoted as following.

Greitzer used Frequency that Stenning defined as follow, (Helmholtz frequency) ω is :

$$\omega = a\sqrt{A_c / V_p * L_c} \quad \text{[Helmholtz frequency resonator]} \qquad (2\text{-}5\text{-}1)$$

The dimensionless mass flow is represented by the axial velocity parameter (C_x / U) i.e.

$$\tilde{\dot{m}}_c = \rho C_x A_c / \rho U = C_x / U \quad \text{[Dimensionless mass flow rate]} \qquad (2\text{-}5\text{-}2)$$

The compressor system compressing through short inlet, the compressor, the discharge line (plenum) and a throttle valve Fig.(2-11) is modeled by the nonlinear lumped parameter equations driven by Greitzer, using symbol (~) to designate the nondimensionlized variables resulting four equations are :

$$d\tilde{\dot{m}}_c / dt = B\,(\tilde{C} - \Delta\tilde{P}) \quad \text{[Momentum balance for the compressor duct]} \qquad (2\text{-}5\text{-}3)$$

$$d\tilde{\dot{m}}_T / dt = (B / G)\,(\Delta\tilde{P} - \tilde{F}) \quad \text{[Momentum balance for the throttle duct]} \qquad (2\text{-}5\text{-}4)$$

$$d\,\Delta\tilde{P} / dt = (\tilde{\dot{m}}_c - \tilde{\dot{m}}_T) / B \quad \text{[Mass balance for the plenum]} \qquad (2\text{-}5\text{-}5)$$

$$d\tilde{C} / dt = (1/\tau)\,(\tilde{C}_{SS} - \tilde{C}) \quad \text{[Relaxation response to departure from steady]} (2\text{-}5\text{-}6)$$

where

$$B = U / 2\omega Lc \quad \text{[Dimensionless parameter equation]} \qquad (2\text{-}5\text{-}7)$$

or

$$B = (U / 2a)\sqrt{V_p / A_c * L_c} \quad \ldots\ldots\ldots\ldots (2\text{-}5\text{-}8)$$

$$G = (L_T * A_c) / (L_c * A_T) \quad \text{[Geometric parameter equation]} \qquad (2\text{-}5\text{-}9)$$

$$F = (\dot{m}_T)^2 / 2\rho (A_T)^2 \quad \text{[Throttle characteristics equation]} \quad (2\text{-}5\text{-}10)$$

Parameter B is proportional to the ratio of the Helmholtz resonator period and the transport time through compressor duct. G parameter is the ratio of equivalent length to

area ratios for throttle and compressor ducts. F is the throttle valve characteristic, and τ is the time lag parameter

τ is proportional to the time for some number N of rotor revolutions, Thus:

$$\tau = 2\pi N_D R / U \qquad \text{[time lag parameter]} \qquad \ldots\ldots\ldots \text{(2-5- 11)}$$

So the nondimensional time lag τ is:

$$\tilde{\tau} = 2\pi N_D R\, \omega / U \qquad \ldots\ldots \text{(2-5 - 12)}$$

$$\tilde{\tau} = (\pi r / L_c)(N / B) \qquad \ldots\ldots \text{(2-5- 13)}$$

$$S = (A_c / A_T)^2 \qquad \text{[Valve characteristic equation]} \qquad \text{(2-5-14)}$$

Where:

a = Speed of sound (mps)
A = Flow area (m^2)
C = Compressor pressure rise
= $\Delta P / (1/2\ \rho U^2)$
C_x = Axial velocity (mps)
F = Throttle pressure drop
m = Mass flow rate
N_D = Time lag in revolutions = 0.5 - 2
ΔP = Plenum pressure rise
R = Impeller tip radius (m)
S = Valve parameter
U = Impeller tip speed (mps)
τ = Dimensionless time = ω t
ω = Helmholtz frequency

For a given compressor since N, R, and Lc are constant, the nondimensional time log is proportional to 1/B. Given the geometry of the system, the steady-state compressor characteristic curve(C_{SS} - m), the valve characteristic (S)and a relaxation time (τ), equations (2-5-3) Through (2-5-6) can be solved by numerical integration starting from specified initial conditions. This done in order to predict the transient

behavior of the compressor system. These equations can be solved numerically for different compressor and throttle characteristic curves, using fourth order predictor corrector in order to determine the dynamic behavior of the compressor system. From the analytical investigations, it found that:

G have only minor effect so it kept constant at experimental value of (0 - 0.36).

B is most critical parameter.

N is assumed to be = 2 by Greitzer and assumed to = 0.5 by Hansen to give better arrangement with his experiment data.

Establishing the theoretical modeling taking in the consideration that:

- Using the Rung-Kutta subroutine *Fig.* (12) to integrate the mentioned four differential equations (2-5-3) to (2-5-6) for m_c, m_t, ΔP, and C.
- Initial values will be:

$\Delta P = C$

$\dot{m}_c = \dot{m}_t = C_x / U$

$N_D = 0.5 - 2$

2.6 Proposed study case modeling: was used this

~~Using~~ Hansen modified model ~~to be~~ applied in study case, because of using the available compressed air instead of centrifugal compressor taking in consideration the following:

- Using calculated C_r from measured flow rate [$Q = (\pi Db)\, \varepsilon\, C_r$] instead of $C_{x\,c}$ of the flow at inlet as Hansen did. The term (πDb) represents the compressor flow area (A_c).
- So the equivalent calculated axial velocity is [$C_r = Q / A\, \varepsilon$ (mps)].
- The resultant head of compression system is a function of pressure coefficient (ϕ)
- The pressure coefficient for turbomachines is $\phi = 2g\, H / U^2$.
- So the equivalent calculated impeller tip velocity is $U = (\sqrt{2g\, H}) / \phi$ (mps)
- But the equivalent calculated velocity is $U = \pi DN / 60$ (mps)
- So the equivalent calculated speed $N = 60 * U / \pi D$ (rpm)
- The characteristic real case performance curve data (dP-Q%) is used instead of (C_{SS} - m) of the system under study for simplicity of modeling.
- The characteristic curve fitting equation is mapped according to curve fitting with polynomial coefficients of fourth order.

Where:

D = impeller tip diameter

ε = Contraction ratio of compressor blades = $\pi D - tz / \pi D$

t = Blade thickness.

Z = Number of blades.

- Using forth order Rung-Kutta integration, the subroutine RUNG will handle the required integration of the four deferential equations (2-5-3 to 2-5-6) and print the result data file for specified case.

It should be noticed that the units of (time may used here are Helmholtz resonator periods ($t/2\pi$). Although the foregoing arguments have focused on the behavior of the simplified compression system, the basic qualitative conclusions can extend to the general case described by Equations (2-5-3) to (2-5 -6). The Hansen's modified based (Greitzer) modeling is enough accurate and the solution offered of solving the four Non Linear differential equations which will be done using forth order Rung Kutta integration to solve these equations for real case offering fast enough computer program which could be easily implemented through expert system used for process monitoring and control strategy.

2-7 Anti *Surge* Control System:

The objective of a compressor *surge* control system is to maintain the flow through the compressor at or slightly above the *surge* line, even through the process demand may be at or below this level. This is accomplished by either blowing off the discharge flow or recirculating it to the compressor suction. A typical recirculation system is shown in *Fig. .(2-6-1)*. A valve in the recirculation line regulates the circulation flow so that the sum of its flow and the flow to the process, that equals the compressor flow is at or above the *surge* line F. This valve called the *surge* control valve, since the blow off or recirculated flow represents wasted power. The control system must not open the control valve when the process demand is above the *surge* line nor should it recirculate more flow than is required. This is to just hold the compressor flow at a *safe level*. Blow off systems used only for gases such as air, which has very little value, and do not create a hazard when released to atmosphere. Normally a *surge* control line established from 1 to 5% above the actual *surge* line to provide for control system overshoot and recovery. Control systems that have good dynamic response, and what commonly called good *start-up* characteristics, permit the *surge* control line to be placed very close to the actual *surge* line.

Information on the actual *surge* line of a centrifugal compressor provided by most manufacturers in a form similar to *Fig.* (2-1),(2-13). The curves shown on such a graph usually based on actual test data and the equation of the actual *surge* line is not available. This means that the determination of the control system computations required to approximate the *surge* control line most easily found graphically.

Data frequently presented as discharge pressure vs flow with the test suction pressure stated so that the data can be replotted as pressure ratio vs flow. On a plot of pressure ratio vs flow for a specific compressor, the *surge* control line can be drawn to provide the desired safe margin between the actual *surge* line and the *surge* control line. These easily have done by picking several points on the actual *surge* line and multiplying the flow value for each point by one plus the safety margin (1 + percentage margin /100) and replotting the new flow value at the same pressure ratio.

2-6-1 *Surge* control system:

The objective of a compressor *surge* control system is to maintain the flow through the compressor at or slightly above the *surge* line even through the process demand may be at or below this level. This accomplished by either blowing off the discharge flow or *recirculating* it to the compressor suction. The typical recirculation system shown in *Fig.*(2-12). A valve in the recirculation line regulates the circulation flow so that the sum of its flow and the flow to the process, which equals the compressor flow, is at or above the *surge* line, this valve called the *surge control valve*. Since the blow off or recirculated flow represents wasted power, the control system must not open the control valve when the process demand is above the *surge* line, nor should it recirculate more flow than is required to just hold the compressor flow at a *safe level*. Blow off systems used only for gases such as air, which has very little value, and do not create a hazard when released to atmosphere. Normally a *surge* control line is

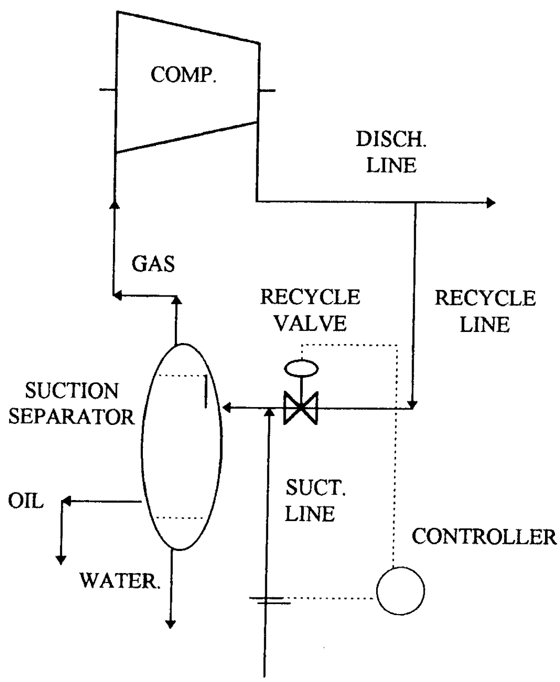


FIG (2-12) GAS COMPRESSOR RECYCLE ROUTINE TO AVOID SURGE [ELLIOT, 1977]

established from 1 to 5% above the actual *surge* line to provide for control system overshoot and recovery.

Control systems that have good dynamic response, and what commonly called good start-up characteristics, permit the *surge* control line to be placed very close to the actual *surge* line. Information on the actual *surge* line of a centrifugal compressor provided by most manufacturers in a form similar to *Fig.* (2.-1). The curves shown on such a graph usually based on actual test data and the equation of the actual *surge* line is not available. This means that the determination of the control system computations required to approximate the *surge* control line most easily found graphically.

Data frequently presented as discharge pressure vs flow with the test suction pressure stated so that the data can be replotted as pressure ratio vs flow. Suction temperature also given since it is an important test parameter and is useful in system design. Plotting the pressure ratio vs flow for a specific compressor, the *surge* control line can be drawn to provide the desired safe margin between the actual *surge* line and the *surge* control line. This is most easily done by picking several points on the actual *surge* line and multiplying the flow value for each point by one plus the safety margin (1 + percentage margin /100) and replotting the new flow value at the same pressure ratio. [Vandaveer, 1982].

In order to determine Losses, Velocities , for *unsteady flow* condition, hence the theoretical analysis in this case is so difficult, but the main decisive factor is the discharge flow rate which can be concluded from *surge* modeling using modified Greitzer and Hansen model.

2.6.2 Application data:

One of the biggest problems in establishing good *surge* control is obtaining a good value for orifice dp signal. Since the piping around most compressors is desirable to keep the recirculation line as short as possible because of installation costs, little or no attention is given to installing orifice in properly designed meter runs. Also the gas flow near the compressor can be extremely turbulent particularly on the discharge side due to compressor design. Therefore, the use of some form of straightening vanes between the compressor and the orifice run have been considered by the system designer to minimize turbulence at the point of measurement. Although damping the primary or output of the transmitter may be satisfactory for some recording purposes, it is not satisfactory in this control system where dynamic response and accuracy are important.

In the best designed gas flow measuring systems, some transmitters primary damping is usually necessary to permit stable control. However, great care must be taken to minimize the damping required or used. This may not be necessary in the actual application of the compressor.

2.6.3 Compressor Anti *surge* controller :

The total analysis of the *surge* control systems discussed here has been based on compressors which are run at variable speeds. One common type of centrifugal compressor is run at fixed speed and inlet guide vanes are adjusted to change the head-flow characteristics. However, this does not alter the approach to determining the best approximation of the *surge* control line since the basic equation of the actual *surge* line does not change.

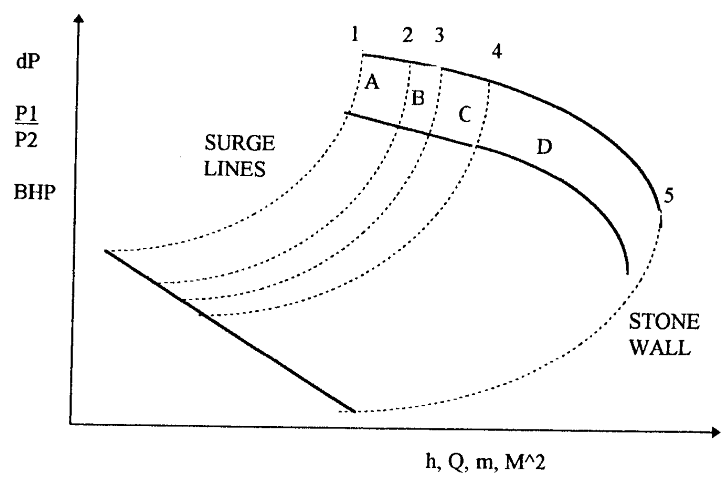


Fig (2-13) ANTI - Surge Control Algorithm [ARAMCO, 1982]

(1) : Actual Surge Line
(2) : Surge Control Line
(3) : Slow Control Line
(4) : Anticipating Control Line
(5) : Stone Wall Line
A : Control Region
B : Control Region
C : Control Region
D : Operating Region

2.6.2.3 discharge valve control:

controller normally controls the discharge pressure of the compressor. In case of the recycle valves opens at a fast rate, dynamic coupling is employed. If the discharge pressure increases beyond the break point then adaptive tuning is used to control the discharge pressure.

2.6.3 Incipient *surge* :

Incipient *surge* is one of the many ways available to detect compressor *surge* . Incipient *surge* is due to slight imperfection in the matching of impellers during compressor design and manufacturing. The compressor suction flow starts oscillating as the compressor approach *surge* . oscillations may also occur when process conditions widely different from the design conditions of the compressor exist. The oscillation amplitude can be determined by running an actual *surge* test and setting an amplitude valve slightly above the normal oscillation amplitude during normal operation .

The amplitude of oscillations near *surge* is so dependent on physical conditions of the compressor such as:

1. Matching of impellers.
2. Design of volute casing.
3. Number of impellers.
4. Physical properties of the gas being compressed.

Some compressors may not exhibit any detectable oscillations until actual *surge* oscillations occur before actual *surge* .

2.6.4 Surge spike:

The elbow taps are not ideal means to measure Incipient *surge*, however, the detection of a *surge* Spike is possible. A *surge* Spike is defined as (the first indication that the compressor is in *surge)*, and that reversal of flow into the compressor is occurring. The spike detection is the last attempt by the controller to prevent actual *surge* . When *surge* spikes are detected by the controller, it opens the recycle valve wider than goes to minimum flow control.

2.6.4 Anti *surge* control:

The following conditions have to be strictly avoided by suitable safety devices [Solzer, 1972]:

- Blade resonance caused by electrical mechanical or aerodynamical excitation can be eliminated by appropriate design measures , where by restriction of the speed range considerably simplifies this task .
- Transient resonance conditions during starting or running out are harmless. *surge* conditions occurring on the left of the *surge* line must be avoided by an effective and rapid Anti - *surge* controller. As soon as the operating point approaches the *surge* line at any point the anti *surge* controller should starts opening the anti *surge* valve thus venting excess air not used by the system to atmosphere.
- Anti - *surge* controller is a protecting device which has to act independent of any other control.
- Rotating stall occurs near the lower part of the *surge* limit (for axial compressors at speeds below about 85% of the nominal speed) operating the compressor under equally dangerous stall conditions are avoided by designing the anti *surge* controller

accordingly because an accidental speed reduction below 85% is thereby rendered harm less to the blading.

2.6.5 Back flow protection :

Zero flow or back flow due to stall conditions at full speed and open or closed check valve immediately brings the air temperature up to near 1000 c as the whole shaft power is used to heat up the air in the compressor. At this temperature the material strength of the blading is reduced to a fraction of its original valve and a complete blade failure would be occur (in blast furnace application) .

A separate protecting device again measures the flow by means of a (Δp) tapping. as soon as this (Δp) falls below a preset minimum, an alarm monitor is switched on and after 5 seconds the Anti *surge* valve is fully opened if in spite of this abnormal condition prevails, the compressor is automatically shut - down 15 seconds later, on order to save the blading.

[patlovany, 1986] analyzed a large air compressor for the purpose of minimizing energy costs and protecting the compressor from *surge*, his analysis shows that *surge* occurs for two different multi stage compressors at the rate of 66 % and 69.5 % of discharge flow at different rotating speeds (from 4280 rpm to 5110 rpm) (*Fig.* 2-14) the plotted *surge* line is shifted positively with gas temperature increases. The method of *recycling as an* Anti - *surge* control is commonly employed where operating pressure levels, inlet temperature, and mol. weight are held fairly constant.

In case of *recycling* cooled hydrocarbon gases through recycle control valve Fig.(2-13) to compressor suction to avoid *surge*, the cooled gas by passed will

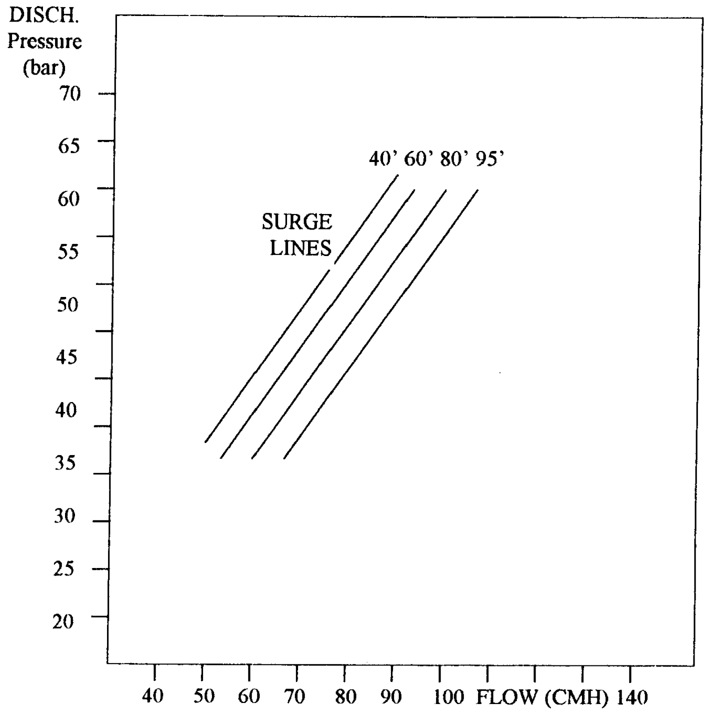


FIG (2-14) SHIFT OF SURGE LINE WITH TEMPERATURE
[PATLOVANI, 1986]

be lighter mol. weight gas which requires higher speed to develop the same head required, for fixed speed driven compressor. The by -passed gas make the compressor may goes deeper into Surge. due to higher head requirement depending on the different of mol. weight.

3.1 Introduction:

In this chapter experimental work will be exhibited in details in four sections. *First* section handles the steady state monitoring and logging of the hydraulic bench as a compression system (test rig) through DAS, the steady state performance curves are obtained. *Second* section handles the unsteady operation monitoring and logging of compression system and exhibit the related performance curves. *Third* section handles the unsteady state modeling of the compression system using the modified Hansen and Greitzer technique discussed in chapter (2), *Fourth section* exhibits the quantitative parameters affects the operation of the compression system to construct a proposed Anti *Surge* Control system as a predictive and preventive control strategy for centrifugal compressors.

3.2 Experimental Apparatus (PMC) System:

the objective of this experimental work is to have a hydraulic process system that has multi-process variables such as (P, T, dp, etc.). As electronic signals (mA, mV, Resistance, On-off pulses, etc.) linking these active signals to a data collecting device called data acquisition system (DAS) that proceed these modulated signals to a personal computer (PC) in order to display and monitor various process variables as a real-time display. the test rig hydraulic process utilized considered a flow bench [Feiersien, 1987]. It is designed to demonstrate the operation of a number of commonly used instruments and methods of flow measurement and provides the

means for their calibration and accuracy comparison. [Validyne, 1987]. Data could acquire in either a manual mode or by means of (DAS).

The flow metering elements are installed in series in a 2.54 cm diameter pipe, mounted on a portable panel Fig.(3.1-1). Compressed air supplied by a separate reciprocating compressor which discharge into a receiver. From the receiver, the air passes through a pressure regulator (V1) and the metering elements:

1. Flow nozzle (FN).
2. Laminar flow element (LFE).
3. Venture meter (VEN).
4. Orifice plate meter(O.P).
5. Turbine meter (TM).
6. ROTA meter (RM.).
7. Bellows Meter (BM).

The measured variables are:

1. Five pressure measuring points.
2. Five temperature measuring points.
3. One differential pressure measuring points.

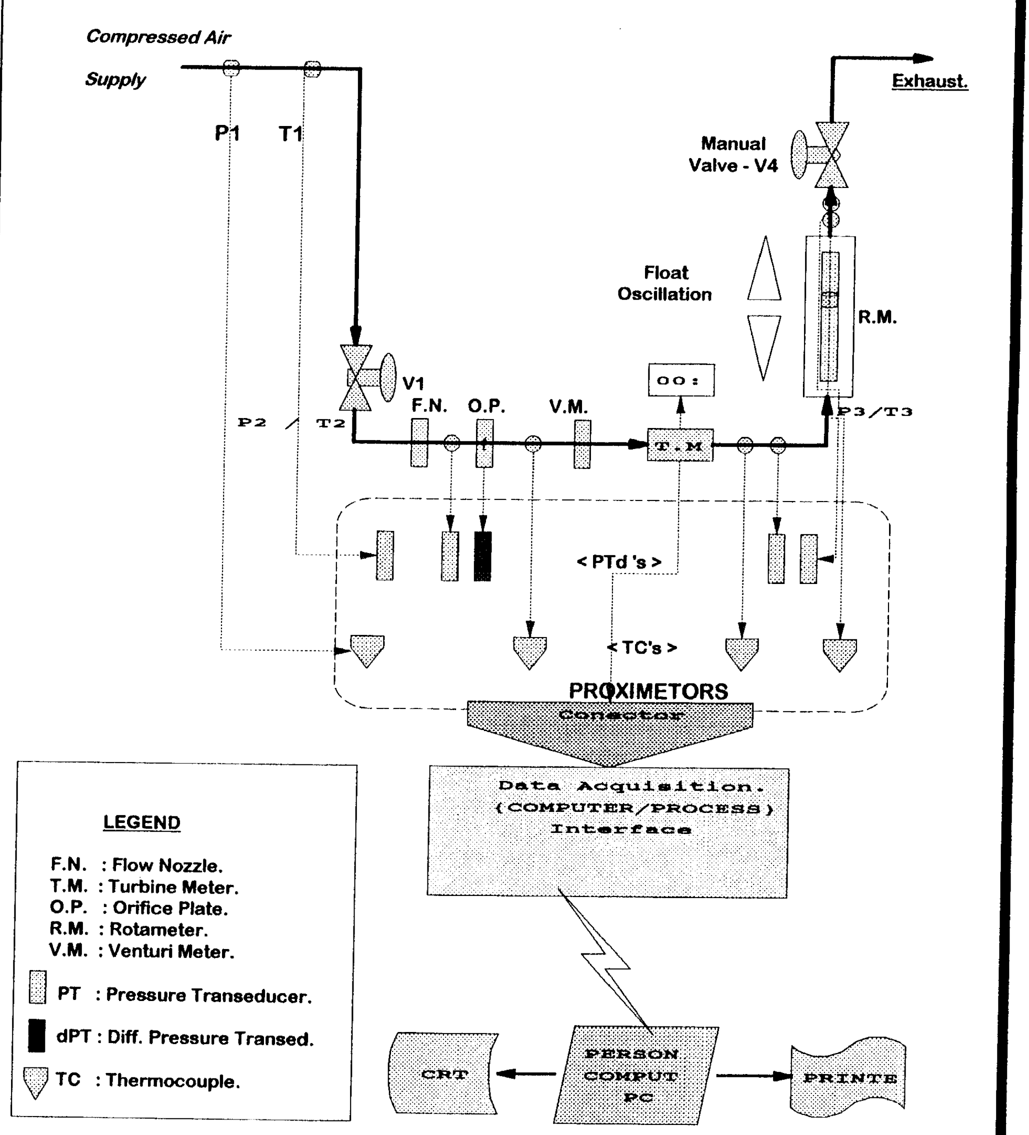


Fig (3-1) Test Rig arrangement {Process Monitoring & Control (PMC)}

From which it discharges to the atmosphere. The pressure in the pipe and flow through the system controlled by the setting of the pressure regulator (V1) and back-pressure valve (V4). Pressures measured locally with borden pressure gauges and manometers, and measured remotely for use with (DAS) by means of at five locations (P1, P2, P3, P4 and P5). Temperatures are sensed with type-(K) thermocouples at five locations (T1, T2, T3, T4 and T5), see fig.(3-2), and registered cyclically on a local mounted digital indicator. The minimum sample scan rate of the device was only one second.

Custom DAS has it's own software which considered a communication utility for the DAS and PC computer used. This software tool establishes serial communication with custom DAS and links it with the PC used to provide Data Acquisition capability.[Helious, 1986] (Helios Computer Front End P/N 834358 Fluke) and (ProLogger HCL P/N 804963) its capabilities are:

- 8 data bits, Baud rate = 9600, linked to serial port (RS-232-C)

After operating the system correctly (air compressor and hydraulic bench) and check all the pressure gauges and flow meter on the panel board, that the operation is normal and no unsteady operation is exist. The DAS set up must be checked and wire connections between DAS connectors and scanner card no. 161 as an analog / digital converter, it is basic and essential card and card no. 162 (thermocouple / DC volts scanner) which is self calibrating analog multiplexed, for 3 input ranges (64 mV, 512 mV, 8 V) the selection of the range will be done through software, plus connector (no. 175: isothermal connector) it has 20 channels (three poles).(Fig. 3-3), (Fig. 3-4)

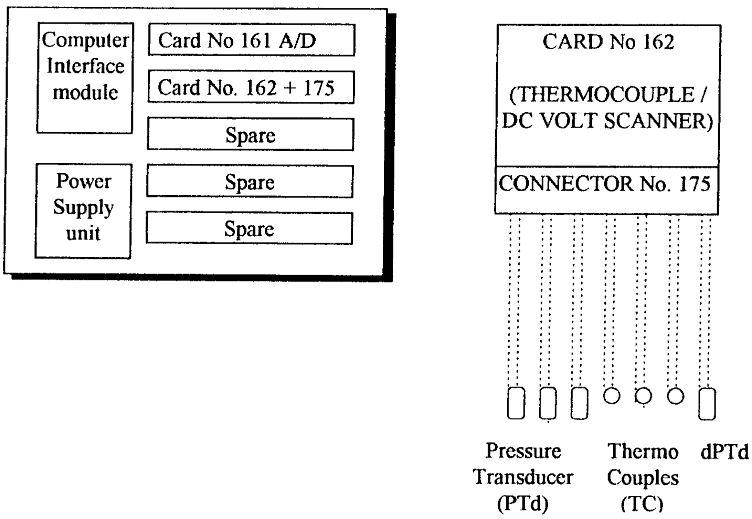
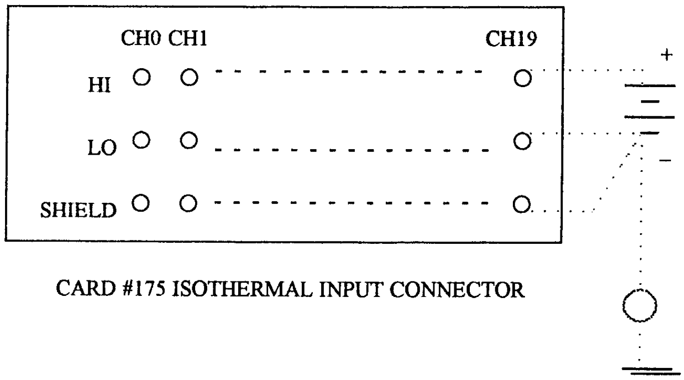
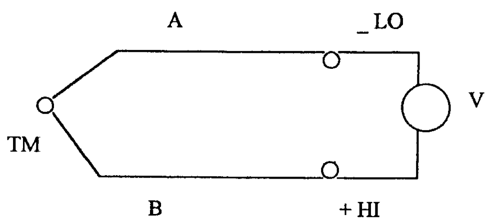


FIG (3-2) DAS (HELIOS) ARRANGEMENT CARDS AND CONNECTORS WITH TC & PTd



(FIG 3-3)
WIRE ARRANGEMENT FOR PRESSURE TRANSDUCER



(FIG 3-4)
TEMPERATURE MEASUREMENTS USING (TC)

3.3 Steady State Analysis:

3.3.1 Computer simulation:

The objective here is to utilize the hydraulic bench described above as a process with multi input variables ready to be linked to the custom DAS for monitoring the steady state operation condition. It's easy to select a mathematical model to apply (simulation and control) procedures for simulation at steady state condition using (computing performance controller), select three inputs signals(MI):

1. (dP) signal across orifice (S1).
2. (P3) pressure signal (S2).
3. (T3) temperature signal (S3).

Using (Computing performance controller) technique [Carlos, 1984] that detailed in (Sec. 3 Ch. 2) and unity scale method to represent a chosen (linearized) equation in order to get a single output signal (SO) that represents the mass flow rate "S4" Fig.(3A-3). This procedure is applied to estimate the density then the mass flow rate .

At steady state condition it was assumed that there are:

P = 101300 N / Mt^2 and T = 300 'k

So from ideal gas law where is:

$$\rho = \mathbf{MP / RT} \qquad \text{.......(3-1)}$$

Using Taylor series expansion mentioned in (ch. 2 sec.3) to get:

$$\rho = \bar{\rho} + \frac{d\rho}{dT}(T - \bar{T}) + \frac{d\rho}{dp}(p - \bar{p}) \qquad \text{...(3-2)}$$

$$\therefore \rho = \frac{M\bar{P}}{R\bar{T}} = \frac{29*101300}{8314*300} = 1.178\ kg/m^{3} \quad ..(3-3)$$

$$\text{and} \quad \frac{d\rho}{dT} = \frac{-\bar{\rho}}{\bar{T}} = \frac{-1.178}{300} = -0.00393\ kg/m^{3}k \quad (3-4)$$

$$\& \quad \frac{d\rho}{dp} = \frac{\bar{\rho}}{\bar{p}} = \frac{1.178}{101300} = 1.163*10^{-5}\ kg/m^{3}-N \quad ..(3-5)$$

$$\rho = 1.178 - 0.00393 * (T-\bar{T}) + 1.163\ Exp\text{-}5 * (P-\bar{P}) \quad ......(3-6)$$

~~Substituting in equation(3-1)~~ to get:

The units are in (SI) units with (KPa and °K or °C) will be in the form of unity scale [Carlos, 1984] as following:

1- Write the equation to be solved along with the range of each process variable (PV)

2- Assign each PV a single name.

3- Relate each PV to its signal name by a *normalized equation.*

4- Substitute the set of normalized-equation into the original equation and solves for the Output signal.

For the test rig process (hydraulic bench) the ranges of the selected operating variables are:

SIGNAL	PV	RANGE	STEADY STATE	SPAN	REM.
S_1	h	0- 100	32.5 cm H2O	100	I / P
S_2	P	0- 1000	400 Kpa	1000	I / P
S_3	T	0- 500	300 'K	500	I / P
S_4	m	0- 20	12.84 Kg / hr	20	0 / P

Table (3-1)

To calculate the mass flow rate of air through the process, a simple equation for the calculation of mass flow through the orifice is :

$$m = K\sqrt{\Delta h\,\rho} \qquad \text{Kg / Sec} \quad \text{(3-7)}$$

Substitute equation(3-6) in equation(3-7) to get that:

$$m = K[h(1.178 - 0.00393(T - \overline{T}) + 1.163 * 10^{-5}(p - \overline{p})]^{1/2} \qquad \text{(3-8)}$$

This equation become the base equation for the custom microprocessor based control system, which has multi input (MI) and single output (SO). That will be a multi input, single output called (MISO) computing performance controller that is similar to Anti - *surge* control system applied for large centrifugal compressors in industry. For the custom hydraulic process orifice coefficient could be calculated by equation:

$$K = m / \sqrt{\rho h} \quad \text{(3-9a)}$$

Where is at Steady State:

$m = 12.84$ Kg /hr and $h = 32.5$ cm H_2O and $\rho = 1.178$ Kg/mt^3

So K= 2.077 ....(3-9b)

The normalized equation used for relating each (PV) to its signal, this means that as the (PV) varies between the low and high values of the range. The signal must vary between the values of 0 and 1, a simple equation to accomplish this:

SIGNAL = Process Variable - low value of range / Span

Applying this equation to the last PV ,(dP, P, T) to get that:

$S_1 = h - 0 / 100 = h / 100$ or $h = 100 * S1$ ....(3-10)

$S_2 = T - 0 / 500 = T / 500$ or $T = 500 * S2$ ....(3-11)

$S_3 = P - 0 / 1000 = P / 1000$ or $P = 1000 * S3$ ....(3-12)

$S_4 = m - 0 / 20 = m / 20$ or $m = 20 * S4$ ....(3-13)

Finally substitute equation (3-9b),(3-10),(3-11),(3-12)and(3-13) in equation (3-8) and solve for the O/P signal (S4) at:

$$\bar{T} = 300\ 'K \quad \text{and} \quad \bar{P} = 400\ \text{Kpa}$$

Then get that:

$$\mathbf{S_4 = 1.035\ [S1\ (\ 2.352 - 1.965 * S2 + 0.0116 * S3\ )\ ]^{1/2}} \quad \text{.............(3-14)}$$

K=1.7%

This final equation can be applied directly to process control system based on microprocessor , and could be also for simulation purposes by varying randomly the values of (I/P): S_1, S_2 and S_3 together to get the (O/P) S_4, on graphic display windows (Fig 3-3-1)

Error analysis :

For steady state condition there are:

$$\bar{P} = 400\ \text{Kpa} \quad \text{and} \quad \bar{T} = 300\ 'k \quad \text{and} \quad \bar{h} = 32.5 \quad \text{cm}\ H_2O$$

So that :

$$\bar{S}_1 = 0.325 \quad \text{and} \quad \bar{S}_2 = 0.60 \quad \text{and} \quad \bar{S}_3 = 0.40$$

Substitute with last mean values in equation (3-14), to get that:

$$S_4 = 1.035\ [\ 0.325\ (\ 2.252 - 1.965 * 0.6 + 0.0116 * 0.4\ )\ ]^{1/2}$$

So that: $S_4 = 0.64$ ....(*)

And from equation (3-13) we find that:

$$\bar{m} = 20 * \bar{S4} = 20 * 0.64 \ = 12.85 \quad \text{Kg / hr} \text{......................(**)}$$

But actual measured STD mass flow rate (By means of Bellows Meter) is:

$$m = 12.84 \quad \text{Kg / hr} \text{..(***)}$$

Then:

Error = ^m / m = 12.85-12.84 / 12.85 = 0.00078 = 0.078 % .... (3-15)

This percentile deviation could consider accepted and satisfied in many cases.

Uncertainty analysis:

Applying uncertainty analysis[Holman,1984] for equation (3-15) will be as following:

$\delta S_4 / \delta S_1 = 1.088$ $\delta S_4 / \delta S_2 = 0.8$ $\delta S_4 / \delta S_3 = 0.62$

$\omega S1 = \pm 0.01$ $\omega S2 = \pm 0.005$ $\omega S3 = \pm 0.001$

$$\omega S_4 = [(\delta S_4 / \delta S_1 \cdot \omega S_4)^2 + (\delta S_4 / \delta S_2 \cdot \omega S_4)^2 + (\delta S_4 / \delta S_3 \cdot \omega S_4)^2]^{1/2} \quad (3\text{-}16)$$

$$\omega S_4 = \pm 0.011 \quad .....(3\text{-}17)$$

Simulating this system using stand alone software program and PC computer, without the Real process nor the DAS. Finally, control could be done by selecting the sample process signals like (P, T, dP). This will represent multi input signals (MI), utilizing unity scale method [Carlos, 1984]. This to represent the chosen linearized equation in order to get a single output (SO) control signal. That represents the mass flow rate (m) subtracting the desired set point from this output signal we can have a correction signal that controls the valve to achieve steady and safe operation. All of the above items could be arranged locally. The STD Soft ware package was unable to represent the all points graphically together, so it was interested to build up a custom package written in (Quick Basic V-4), programming language. Displaying the custom process graphically with different windows crisp and clear to the user with main menu to select different group displays either graphic, tables, trends, or report which will be a sample package that could be run as a stand alone simulator and could be applicable

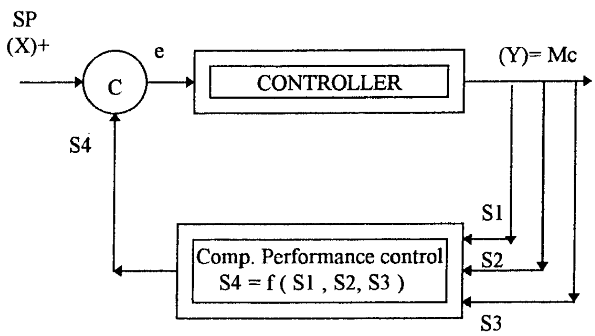
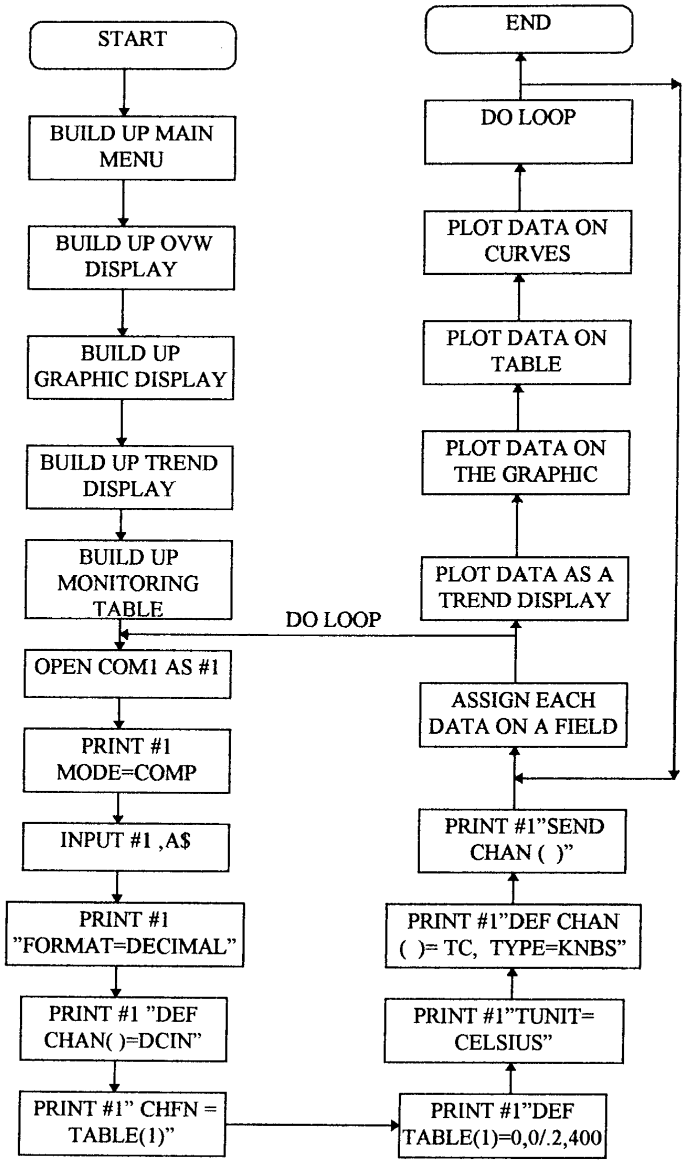


Fig (3-5)
Computing Performance Controller for HYDRAULIC LAB.



(FIG 3-6) PROCESS MONITORING SOFTWARE PACKAGE

3.3.2 Steady state experimental monitoring and logging:

Data for monitoring of the measuring elements and demonstration of their operation can be obtained by operating the system over a range of flow rates and pressures at ambient temperature. This is accomplished by making a series of test runs with varying inlet pressure (P1) and back pressure (P3) these pressures are adjusted with the pressure regulator (V1) and back pressure valve(V4).

- Pressure transducers (variable reluctance type) used provides linear output of (10 V DC) signal with range from 0.5 KPa to 22400 KPa at accuracy of 0.25 % of pressure with output of 1.0 VDC for 2217 Kpa.
- The pressurizing system varies form [0-400] Kpa with transducer output range of [0 - 200 mV].
- Air flow through the system as a reference flow calibrated with bellows meter was Qr. = 131 LPM or Mr = 0.214 Kg/min.
- Turbine meter (TM) coefficient was C = 0.969.
- Orifice plate meter (OP) discharge coefficient was C = 0.604.
- Thermocouple (TC) used was type (K) as National bureau of STD (NBS).

1 kg / sq. cm = 10 Exp. 5 Pa = 10 Exp. 5 N/sq. m = 1000 KPa

Performance curves represents the data trends taken during the series of test runs illustrated by plotting the following curves:

Fig(3-8) as a start up performance curve { P1- Flow Q (LPM), exhibits the system performance during start up at atm. pressure up to 400 Kpa pressure against the flow rate represented at LPM. it shows that flow is increased as regulator valve (V1) (which is considered as compressor discharge control valve) opened to increase the pressure interred to the system, the max. flow is at max. inlet pressure in the system .

Fig(3-9) as a start up performance curve (P1/P2 - Flow Q%), exhibits the system performance during start up, showing the relation between the pressure ratio of P1/P2 against flow rate Q%, the max. pressure ratio is (2.8) which occurs at flow rate of (62.8%) of STD flow rate (0.214 Kg/min).

Fig(3-10): throttling performance curve {P1- Flow Q (LPM)}, where is the Throttling process of the compression system from the max. flow rate at max. inlet pressure to the system(discharge pressure of the compressor), until the system is depressurized completely.

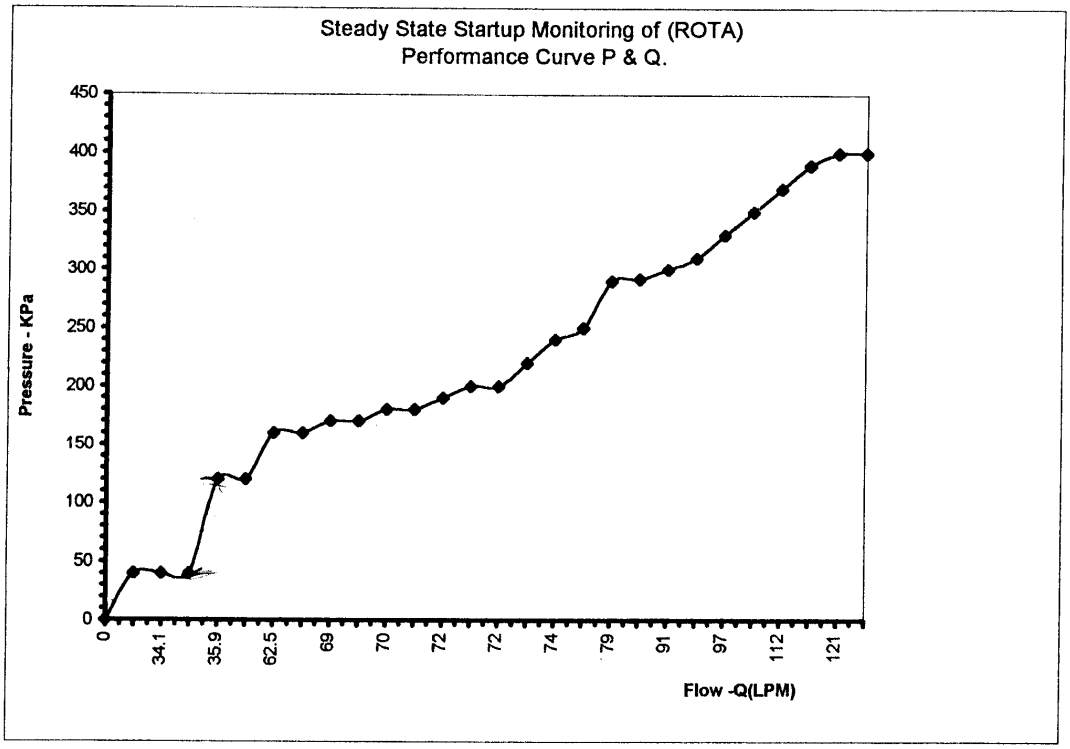


FIG (3 -8) START UP PERFORMANCE CURVE { P1- Flow Q }

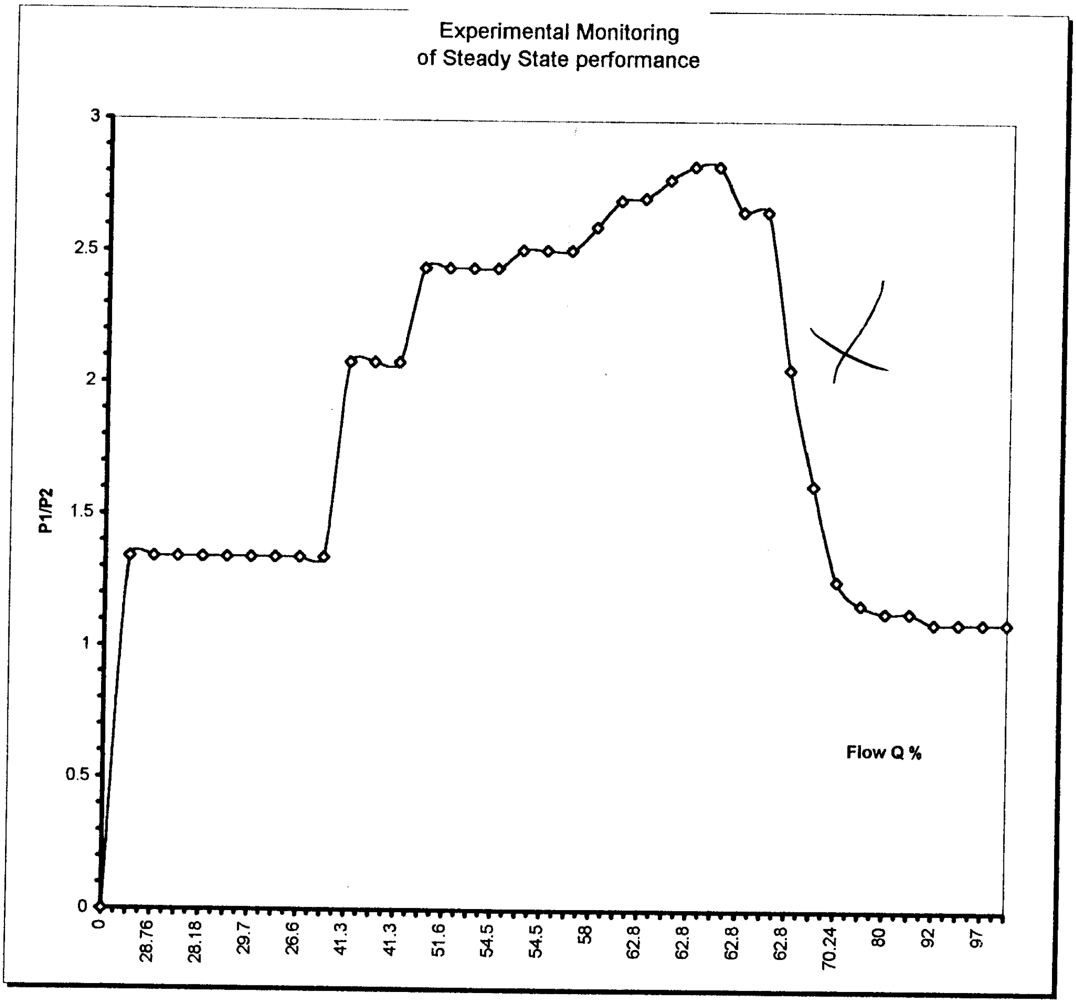


FIG (3 -9) START UP PERFORMANCE CURVE (P1/P2 - Flow Q%)

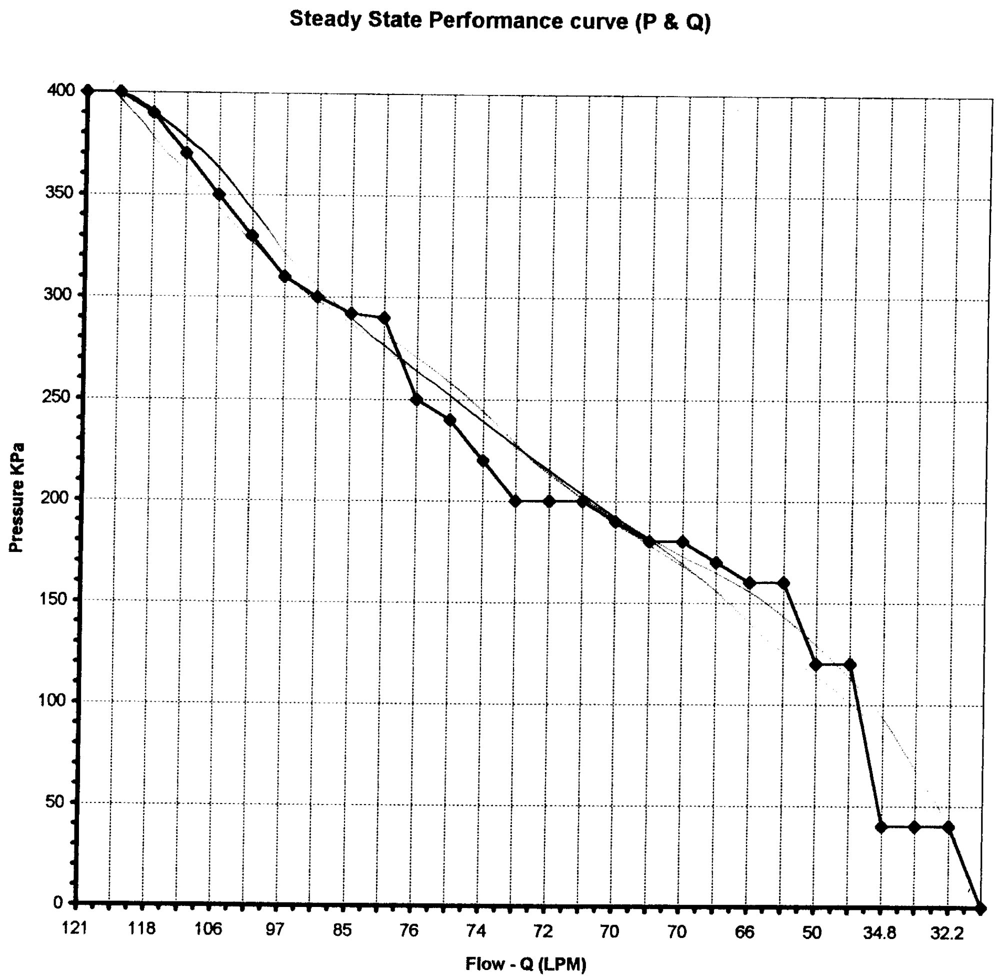


FIG (3 -10) THROTTLING PERFORMANCE CURVE { P1- Flow Q (LPM)}

3.4 Transient Operation Analysis:

3.4.1 Computer simulation

The Hansen modified model was used, the modification is based mainly to expand the modeling in order to handle different geometric conditions and characteristics and different system performance. Using numerical analysis {non-linear Deferential equations (fourth order Rung Kutta integration)}to solve the four Greitzer's equations, as a Fortran executable file (Fig 3-12 &3-13) driven by an Expert System Simulation package then the Modeling program produces a (Test. dat) file including all initial parameters essential for modeling, plotting these parameters against each other to get the required charts in unsteady operation analysis (transient compression system), *Rota* surge analyses is shown in Fig (3-12) and Fig (3-13).

A) Hansen's case modeling:

It is essential to simulate Hansen's modeling case, based on the geometric data of his model and its performance curves stated on his paper. The theoretical modeling of both Hansen's cases were solved successfully and they were typically identical to his experimental models exhibited on his paper. His first case was operating at (speed of N=30,000 rpm & B = 0.55 & S = 125). His second case was operating at (speed of N=54,000 rpm & B = 0.98 & S = 250), where $\{S = (A_c/A_t)^2\}$ is the valve parameter, (Fig. 3-14) up to (Fig. 3-18) shows both cases theoretical modeling at the above conditions.

FIG (3-12) FLOW CHART OF SUB ROUTINE RUNG

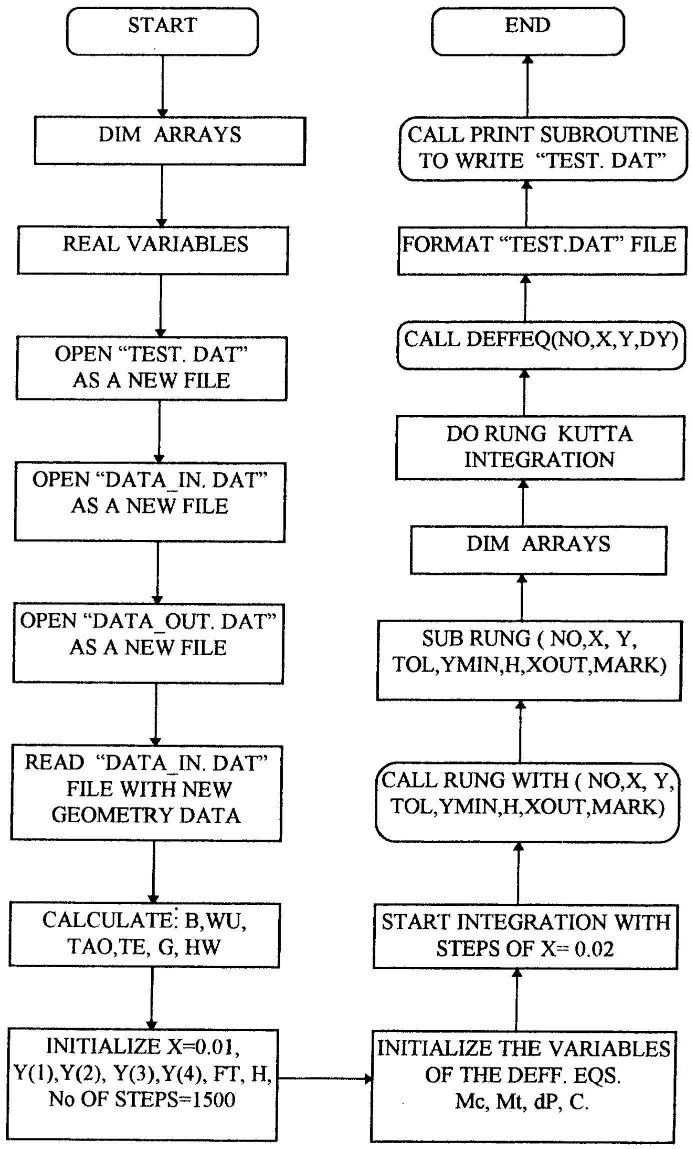
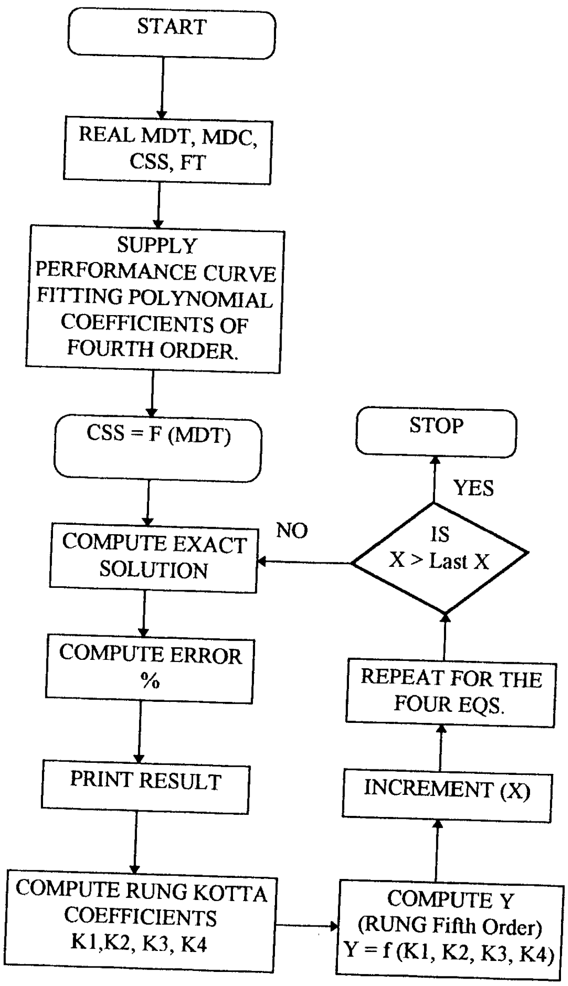


Fig (3-13) FLOW CHART OF SUBROUTINE DEFFEQ.



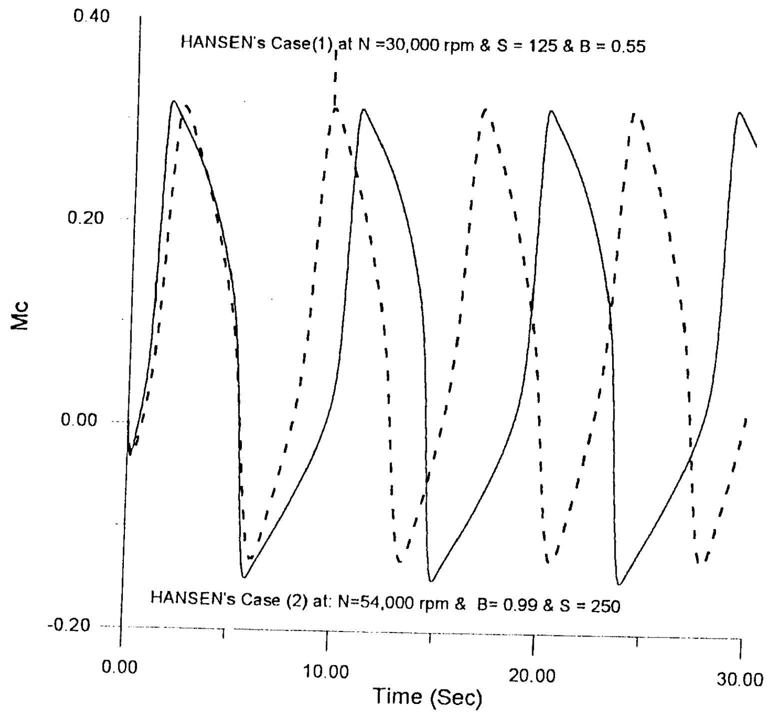


Fig.(3-14) Hansen transient response modeling for his both cases (Mc-Time) Chart

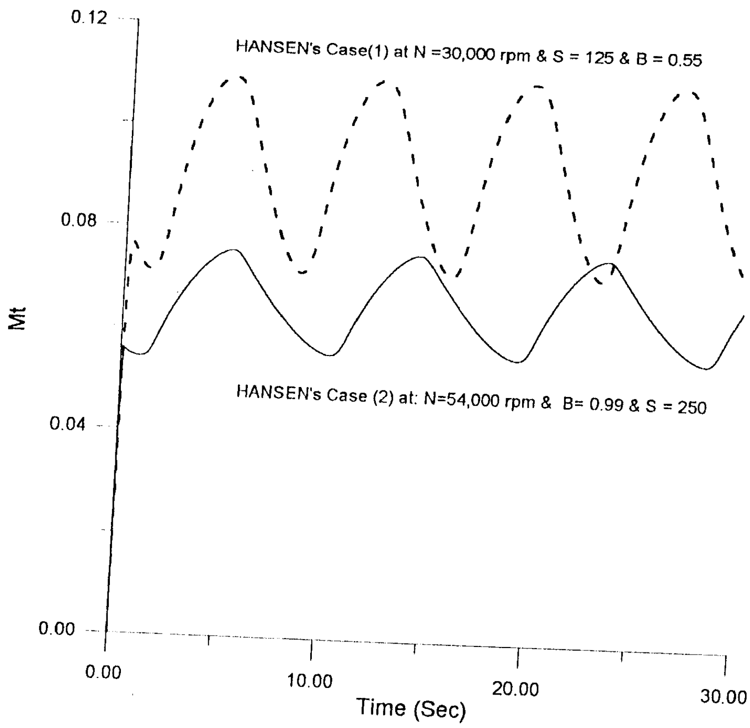


Fig.(3-15) Hansen transient response modeling for his both cases (Mt-Time) Chart

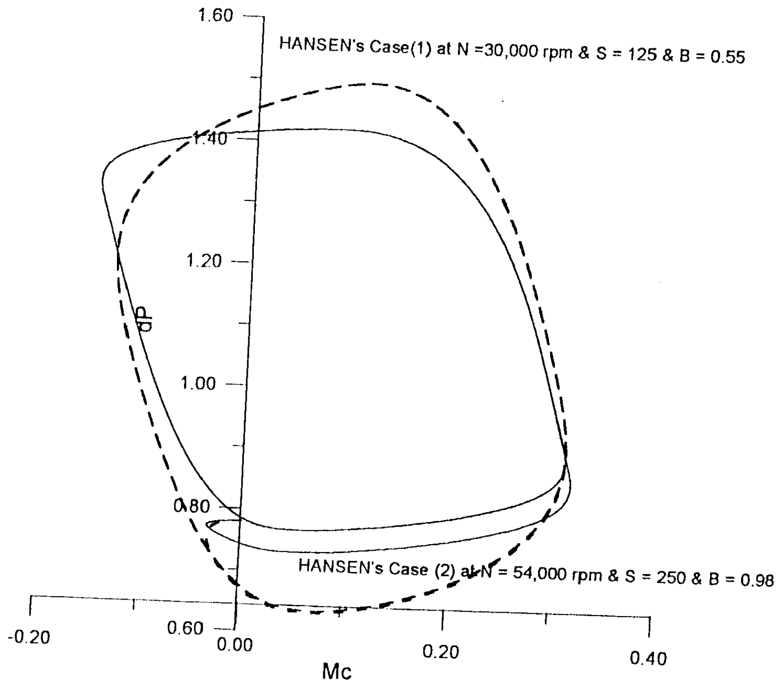


Fig.(3-16) Hansen transient response
modeling for his both cases
(dP-Mc) Chart

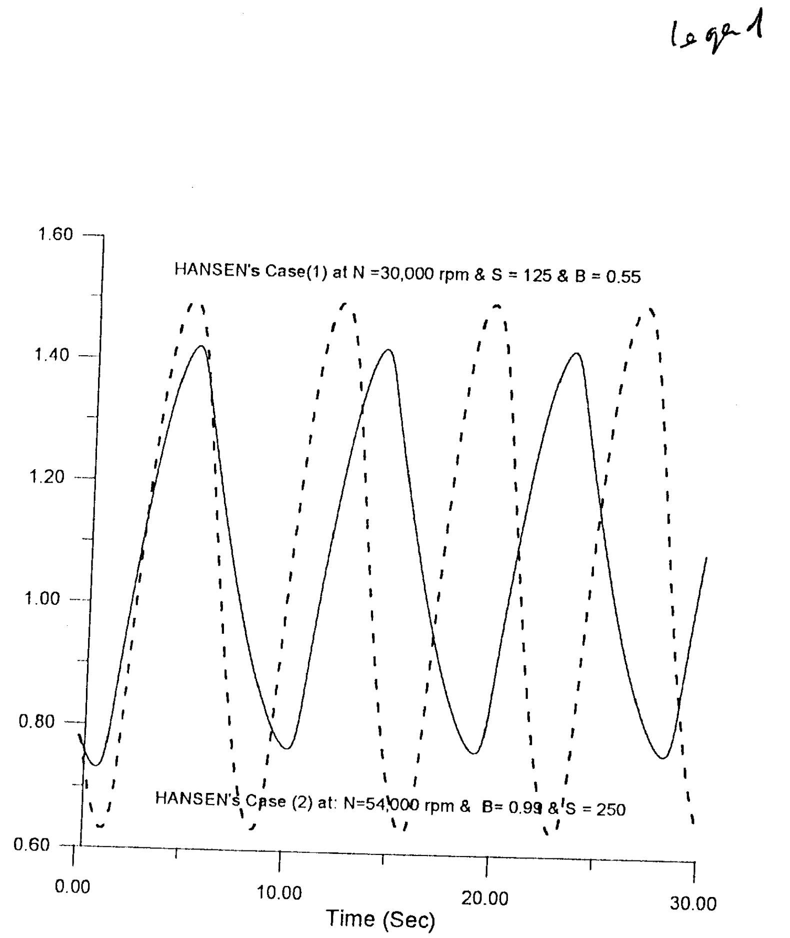


Fig.(3-17) Hansen transient response modeling for his both cases (dP-Time) Chart

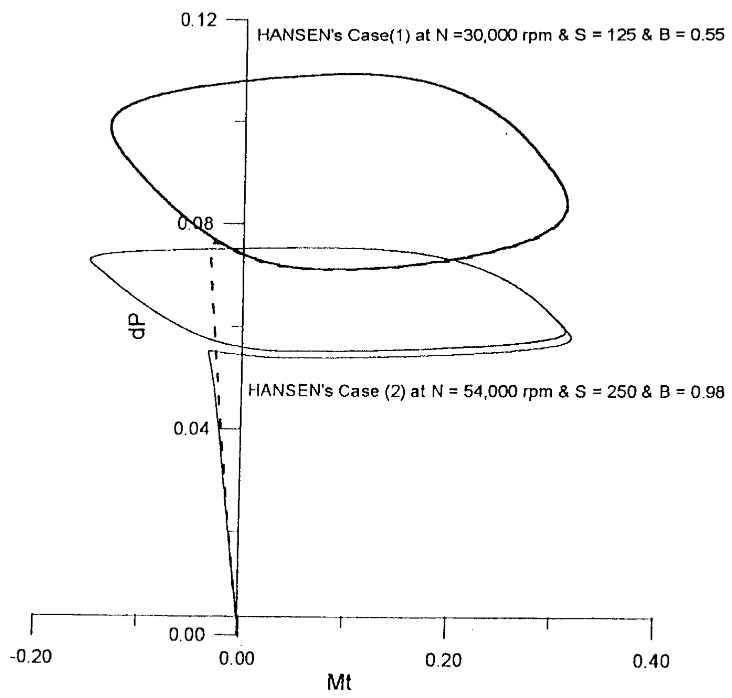


Fig.(3-18) Hansen transient response
modeling for his both cases
(dP-Mt) Chart

B) *Rota* case modeling:

Referring to fig (3-19) there are the *Rota* model arrangement and geometric data of the experiment study model (compression system) as a non linear lumped parameter model. This data will be fed to the modeling program to get the model simulation which will indicate the model instability and surge occurrence at certain parameter values which will be discussed latter.

Test rig *Rota* system was simulated and modeled at three cases (different S-values) enabling to change initial geometric data entry and the polynomial coefficients of the performance curve fitting curve (pr- Q%) fig (3-20) of the system studied so the steady state pressure rise (Css) was mapped in the formula:

$$Pr = 7.2376 - 0.6356 * Q + 0.02257 * Q^2 - 0.0003 * Q^3 \quad ....(3\text{-}4\text{-}1)$$

Compressor Stability (Greitzer)'s parameter at 10930 rpm

$B = U / 2\,\omega L_c = \quad 0.256 \qquad (< \text{critical value})$

The period of surge oscillation for time lag N_D=0.5 rev. is:

$\tau = 2\,\pi R N_D / U \qquad = \quad 0.027 \quad \text{sec.}$

The period of surge oscillation for time lag N_D=2 rev. is:

$\tau = 2\,\pi R N_D / U \qquad = \quad 0.11 \quad \text{sec.}$

Frequency $\omega = a \sqrt{A_c / V_p * L_c} \qquad = \quad 55.57$

$\omega / 2\,\pi \qquad = \quad 8.87 \quad \text{Hz}$

Note that: (Surge cycle = 3 - 10 Hz)

The geometric parameter is

$G = L_t\ A_c / L_c\ A_t = 0.812 \qquad < 1$

Hansen's valve parameter:

$S = (A_c / A_t)^2 \qquad = \quad 125$ (First case)

$= \quad 250$ (Second case)

$= \quad 50$ (Third case)

See performance curve fitting (Pr-Q% Curve) *Fig.* (3-20)

$M_c = C_r / U = 0.69$ (at 100 % Flow and N=10930 rpm)

There are the modeling charts that explains the unsteady operation condition utilizing the Rung-kutta fourth order Fortran program, plotted from test.dat by excel software package.

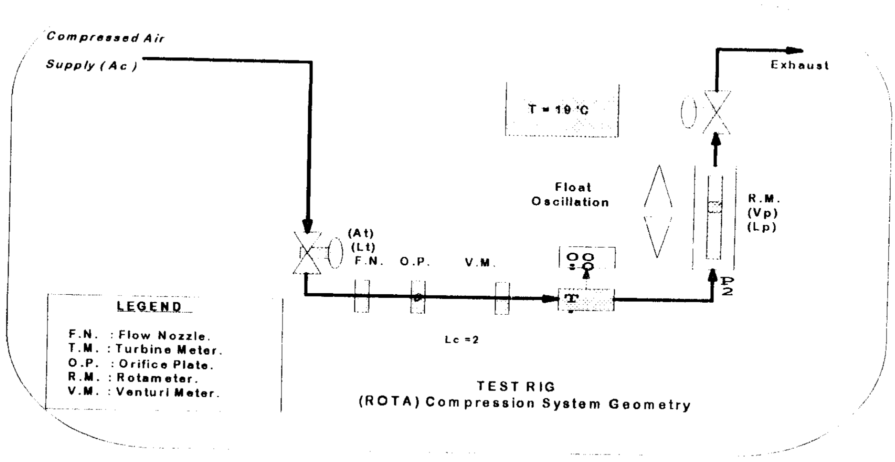
Fig. (3-21) showing the M_c - Time chart where the mass flow *Rota* is surging with time and the surge dissipate gradually according to the initial compression system performance curve (Fig 3-20). The three curves for different (S) values are identical.

Also Fig(3-22) shows the M_t -Time chart which reflects the system behavior as last curve of Mc -Time which enhance the relation between flow *Rota* and valve parameter (S). Also the three curves for different (S) values are identical.

Here in Fig (3-23) is the dP- M_c chart for the same case, this chart reflects the unsteady behavior of the system, the flow starts at the center of the whirl surging hardly and then begin to relax at the end of the whirl lump as the steady state performance curve show before at fig (3-20), note the departure of the curve in the negative region of M_c indicating the occurrence of the reverse flow during surge onset which is the main evidence of surge onset in this case. Also the three curves for different (S) values are identical.

Fig (3-24) Shows the dP- Time chart, that easily exhibits the fluctuation of the pressure ratio against the time indicating the surge onset at this transient operation. Also the three curves for different (S) values are identical.

Fig. (3-25) Shows the dP - M_t chart, that represents the pressure ratio against flow through valve, indicating the importance of the valve setting. Also the three curves for different (S) values are identical.



INITIAL DATA ENTRY

Plenum volume	$V_p = 0.0096$	mt^3
Flow area of compressor	$A_c = 0.0005$	mt^2
Compressor duct length	$L_c = 2.0$	mt
Flow area of throttle	$A_t = 0.00078$	mt^2
Throttle Length	$L_t = 0.25$	mt
Comp. impeller radius	$R = 0.05$	m t
Delay time	$ND = 0.5$	~
Corresponding speed.	$N = 10930$	rpm
Valve Parameter	$S_1 = 125$	~
	$S_2 = 250$	~
	$S_t = 50$	~
Plenum length	$L_p = 0.305$	mt
Speed of sound	$SA = 342$	mps

Fig (3-19) Test Rig Compression System
Geometric Data

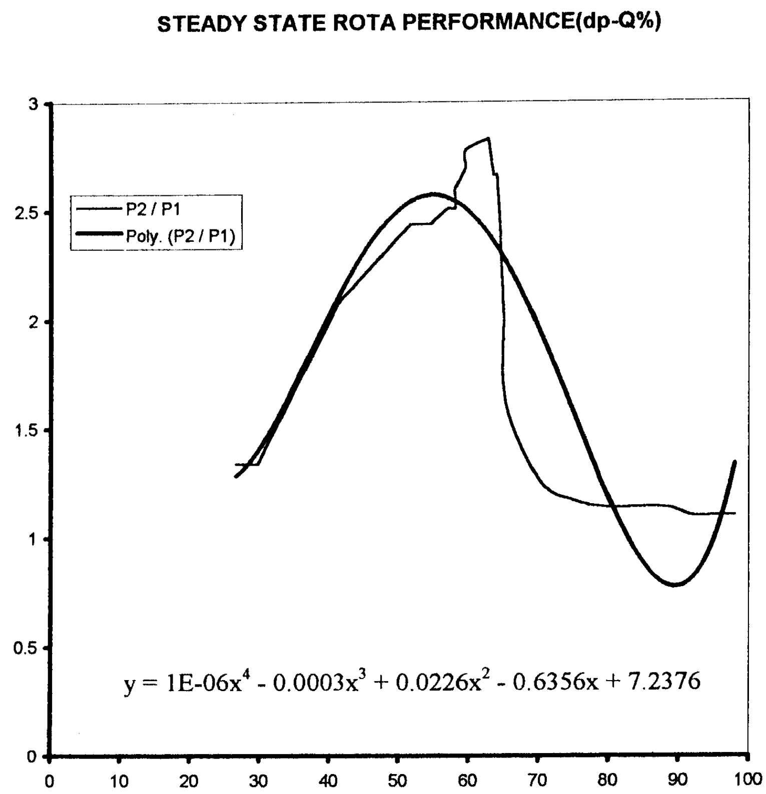


FIG (3-20) STEADY STATE PERFORMANCE CURVE (Pr-Q%)and FITTING CURVE WITH ITS POLYNOMIAL COEFFICIENTS.

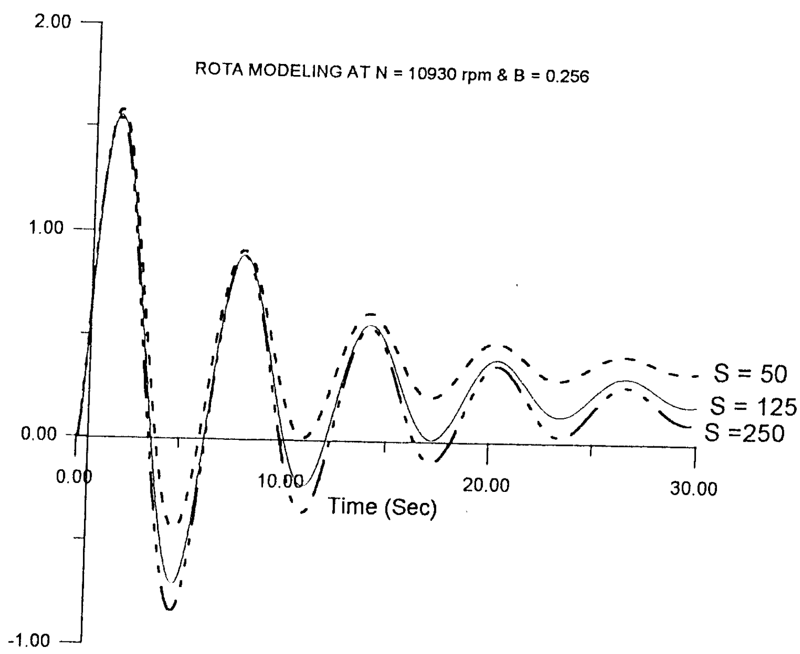


Fig (3-21) TEST RIG (ROTA) MODELING Charts at *Unsteady* operation condition using RUNG-KUTTA fifth order FORTRAN Equation @ B=0256 & N = 10930 rpm and Performance Curve (dP- Q%) Fit Curve.
(Mc - Time) Chart

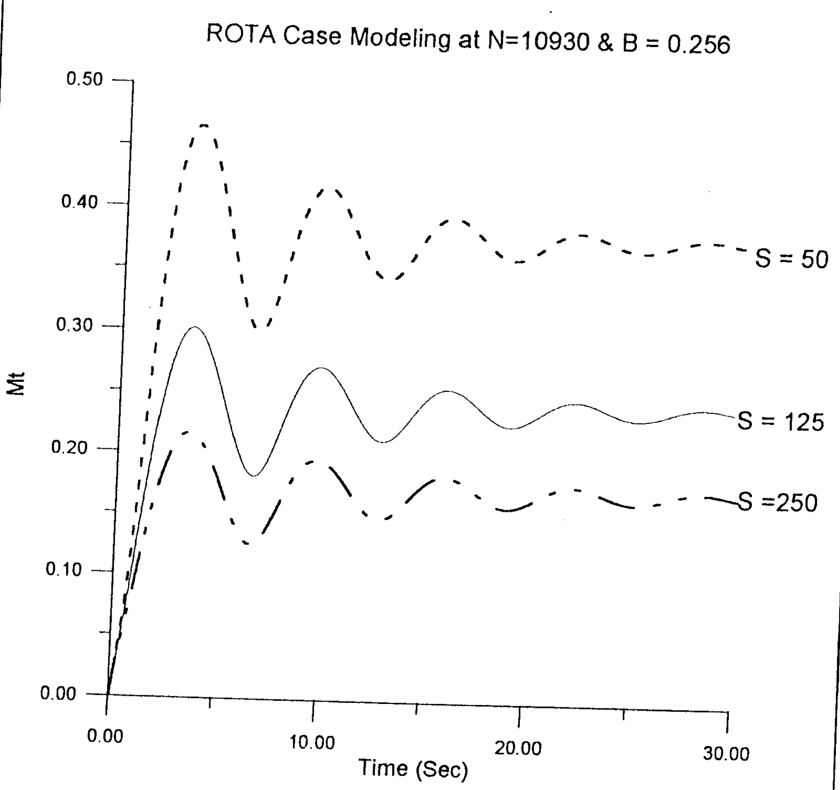


Fig (3-22) TEST RIG (ROTA) MODELING Charts at *Unsteady* operation condition using RUNG-KUTTA fifth order FORTRAN Equation @ B=0.256 & N = 10930 rpm and Performance Curve (dP- Q%) Fit Curve.
(Mt - Time) Chart

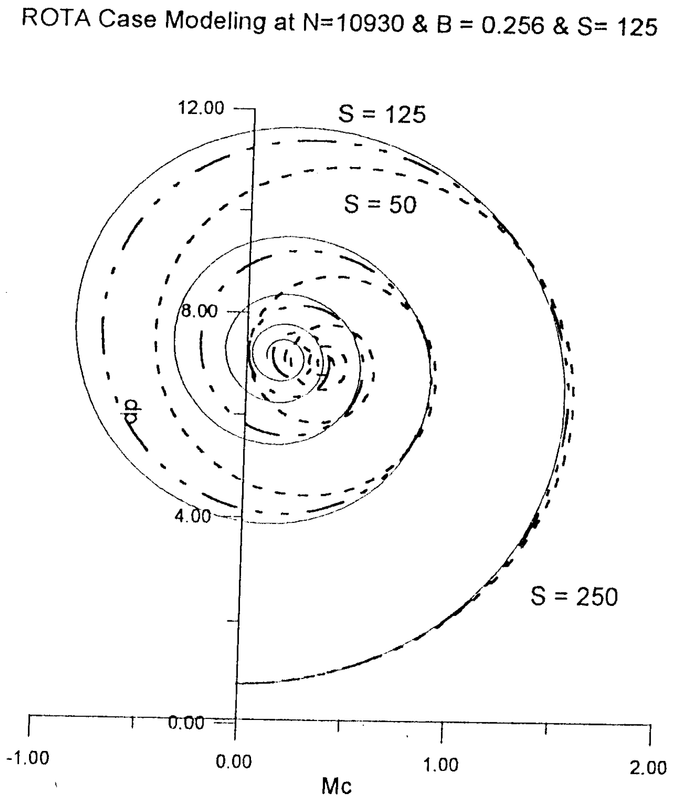


Fig (3-23) TEST RIG (ROTA) MODELING Charts at *Unsteady* operation condition .
using RUNG-KUTTA fifth order FORTRAN Equation @ B=0.256
& N = 10930 rpm and Performance Curve (dP- Q%) Fit Curve.
(dP - Mc) Chart

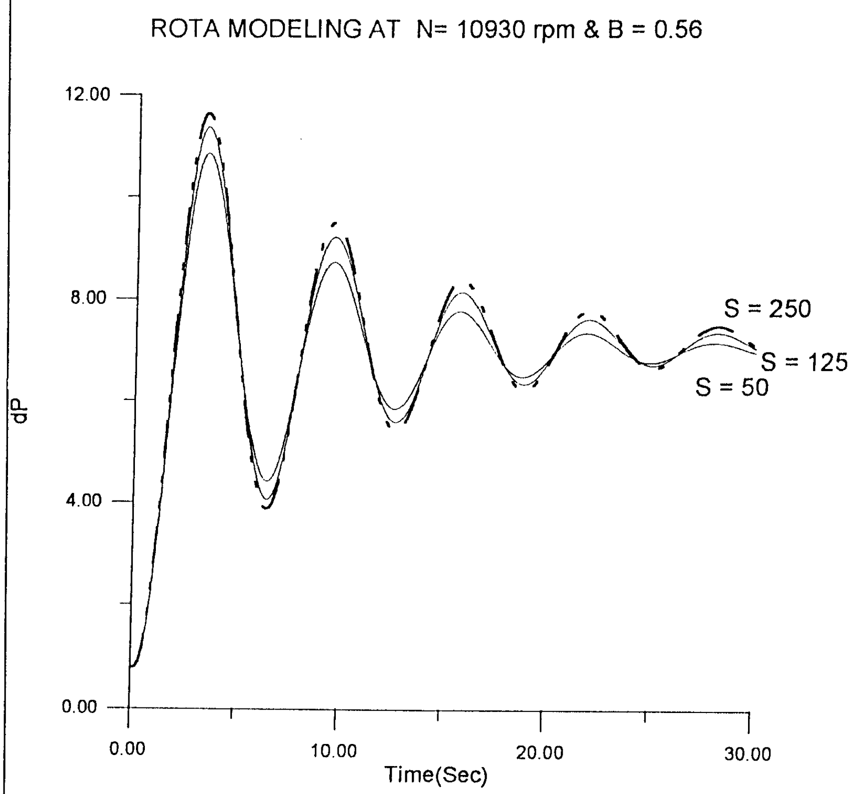


Fig (3-24) TEST RIG (ROTA) MODELING Charts at *Unsteady* operation condition .
using RUNG-KUTTA fifth order FORTRAN Equation @ B=0.256
& N = 10930 rpm and Performance Curve (dP- Q%) Fit Curve.
(dP - Time) Chart

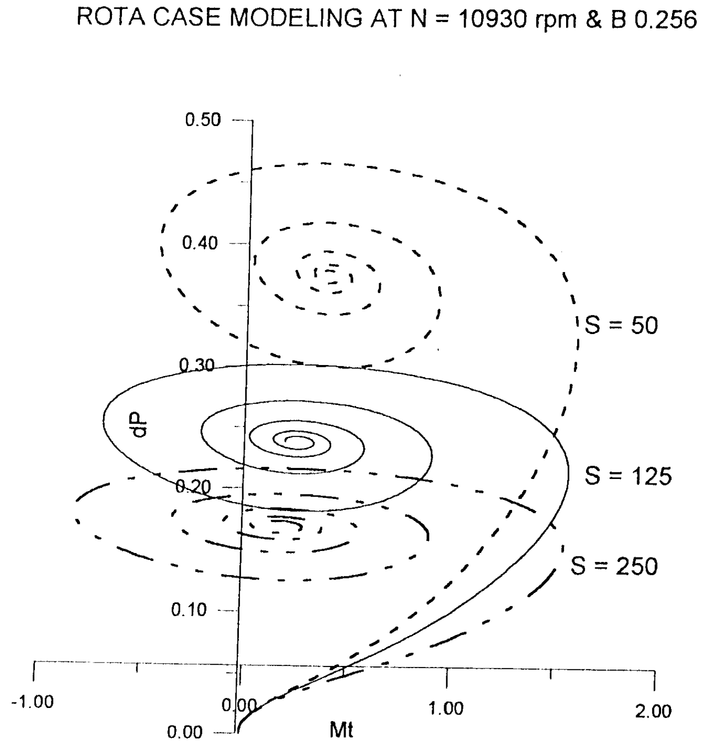


Fig (3-25) TEST RIG (ROTA) MODELING Charts at *Unsteady* operation condition .
using RUNG-KUTTA fifth order FORTRAN Equation @ B=0.256
& N = 10930 rpm and Performance Curve (dP- Q%) Fit Curve.
(dP - Mt) Chart

3.4.2 Experimental results:

During start- up at low flow rate and high pressure ratios it is noticed that due to the large losses resulted from obstruction devices on the discharge line as in fig (3-1) , a clear visual and audible evidence are easily observed. This means that the *float (pop)* of the *Rotameter* as a flow measuring device was surging hardly until the flow rate increased and excess the critical value. During experimental logging it was found that the sample rate of the DAS used was too much slow it was only 1 second which cannot enabling to log the *surge* onset that was less than 1 second, so to complete the unsteady experimental logging, the turbine meter (TM) with it's digital output (LCD)screen was utilized instead of DAS, and manual step by step regulator valve (V1) control was followed and tracing of fluctuation of flow rate in (LPM) reading on the TM catching low and high reading values to be plotted against time, then change the valve (V1) opening to get another reading and so on, noting that the turbine flow meter coefficient was C = 0.969 of the full range, (the sensitivity is

+1%, and the repeatability is + 1%), the surging time rate was 40 *surges* per 20 seconds this means two *surge* onset per second. The *surging* of the float of the *Rotameter* was so interested for monitoring and logging to analyze this phenomenon and correlate it against Hansen's and Greitzer's analysis in the next section.

3.4.2.1 Rota system monitoring and logging at transient condition:

The monitoring of the *surge* compression system at unsteady condition which occur at partially opening of exhaust valve (V2) had been done at several cases as follow:

Fig (3-26); this curve is plotted while an audible and visual *surge* onset on *Rota bob* causing the curve of Flow (Q) against time to be as a sin curve at a fixed valve (V1) opening setting.(Flow uncertainty is +2%)

Fig (3-27); Shows the operating the system over a range of regulating valve (V1) setting while keeping the exhaust valve (V2) partially opened so performance curves fig (3-20) exhibit the system performance of pressure ratio (Pr) against flow rate (Q%), the flow was surging hardly while start up until the flow across the value of 63 % of STD flow rate. (Flow uncertainty is +2%)

Fig (3-28); Here it is interested to plot pressure ratio (P1/P2) against the (S) Valve parameter {S = (Ac / At)^2}to note that the system behavior here is similar to the last curve, this means that the (S) parameter is a quantitative parameter to represent the flow through the compression system [uncertainty of (S) is +5%]

Fig (3-29); In this chart the plotting of the unstable flow which starts to *surge* at low flow rates and surging becomes strong and hard until the flow passed the flow percentage of 63 %, then the flow becomes steady at the rest of the range, the flow represented here against time (Sec.) (Flow uncertainty is +2%)

Fig (3-30) Shows that the flow (Q%) is plotted here against pressure ratio (Pr), the curve is similar to be as last curve notice that the surging occurs at the range of high pressure ratios and the steady flow resumes again at low pressure ratio of about (2.06).

Fig (3-31); shows the plotting of the (S) valve parameter {S = (Ac / At)^2} against the flow(Q%) , it is found that the surging progress while (S) value is high starting with max. (S) value at valve (V1) closing sate, until the (S) value decreases to

equal value of 105 , which is associated with value of 63 % of flow rate and pressure ratio (Pr) at value of 2.8.

Fig (3-32); Shows the plotting of the (S) valve parameter $\{S = (A_c / A_t)^2\}$ together with flow (Q%) against time (Sec.) shows clearly the direct relation between the valve (V1) setting and the flow through the compression system, that affect on the *surge* onset.

Fig. (3-33) is the summary of most important monitoring curves at transient condition of the compression *Rota* system .

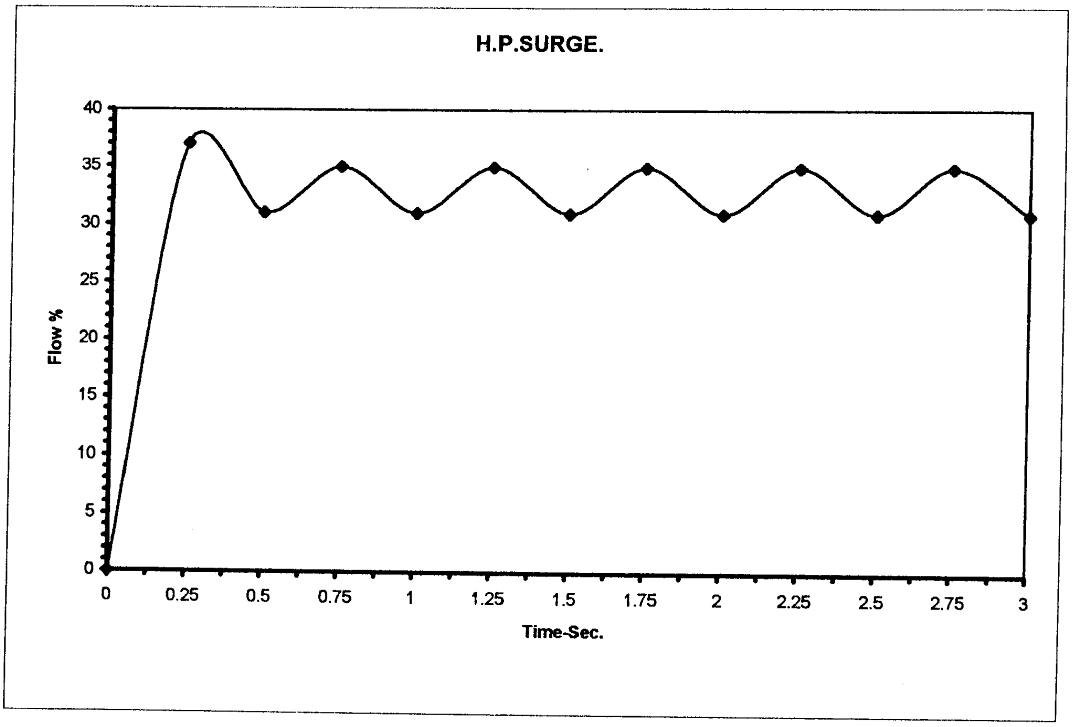


FIG (3-26) EXPERIMENTAL MONITORING AT TRANSIENT OPERATION
(SURGE ONSET)

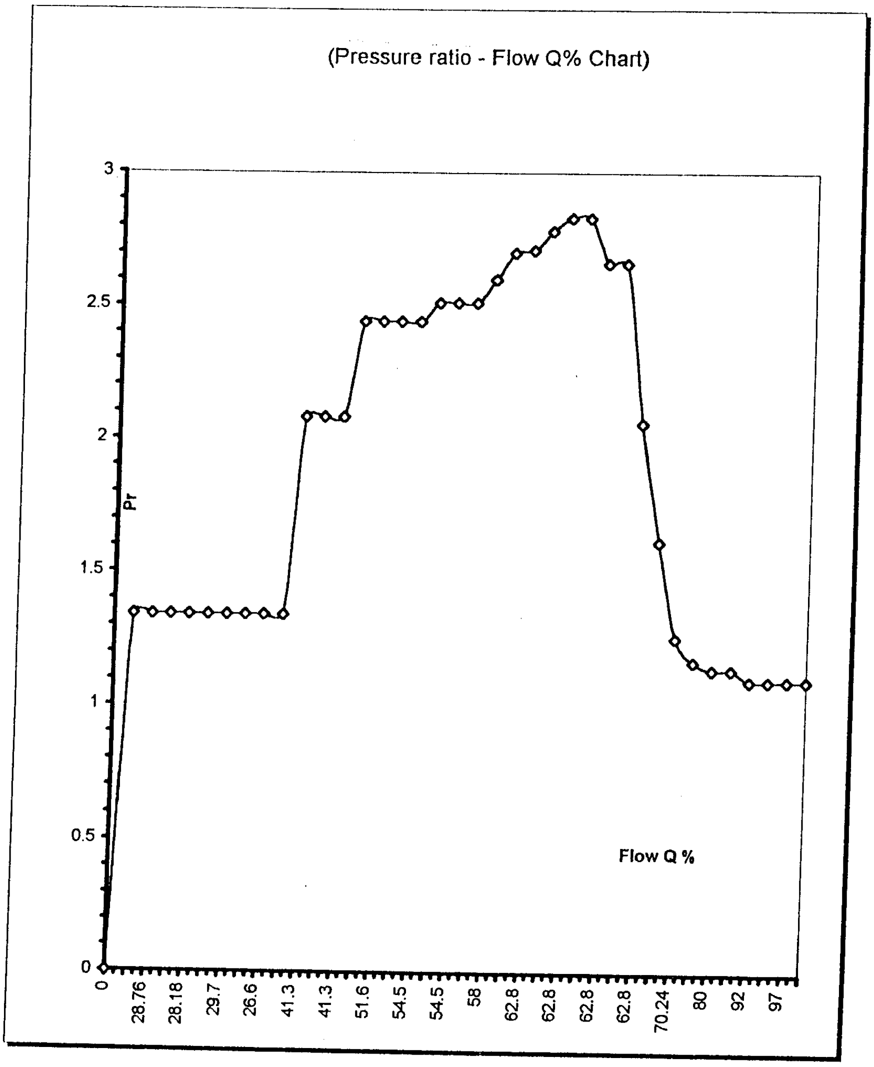


FIG (3-27) EXPERIMENTAL MONITORING OF ROTA COMPRESSION SYSTEM FOR: PRESSURE RATIO (Pr-FLOW Q%) CHART

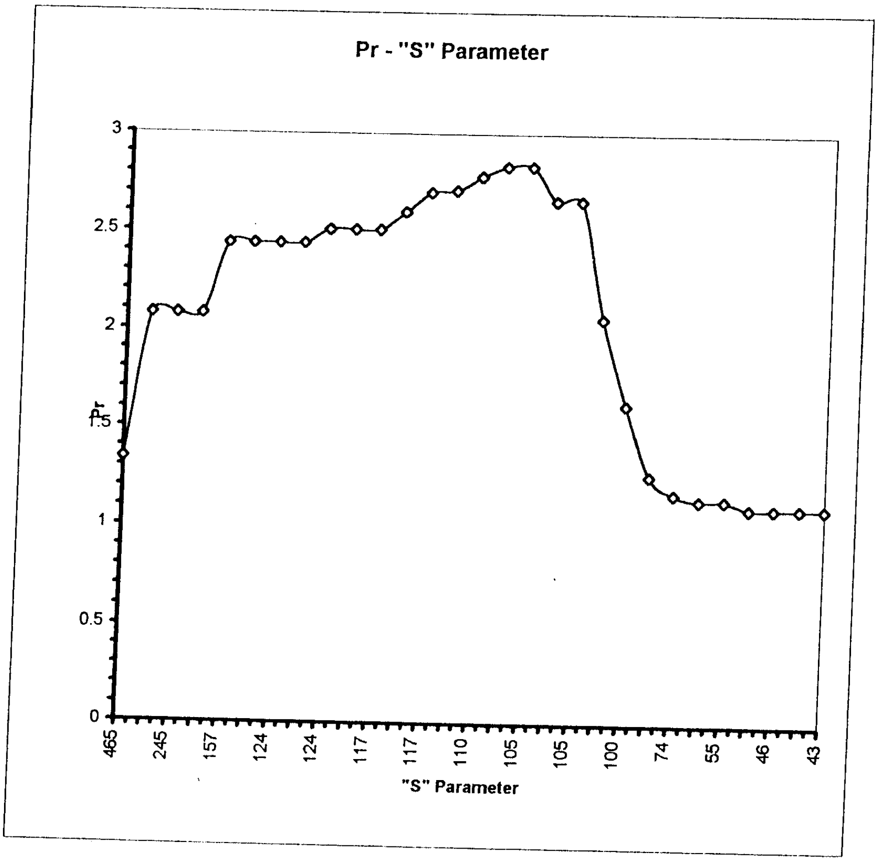


FIG (3-28) EXPERIMENTAL MONITORING OF ROTA COMPRESSION SYSTEM FOR PRESSURE RATIO (Pr - (S) PARAMETER) CHART

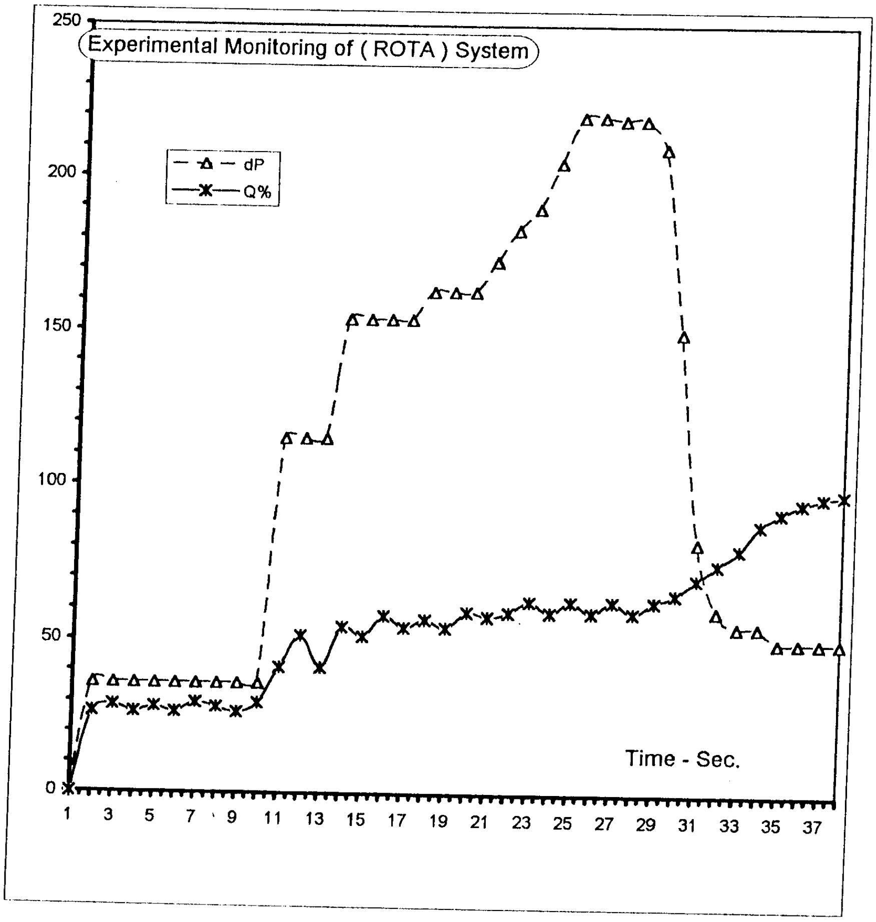


FIG (3-29) EXPERIMENTAL MONITORING OF ROTA COMPRESSION SYSTEM FOR (FLOW Q% -TIME Sec) CHART.

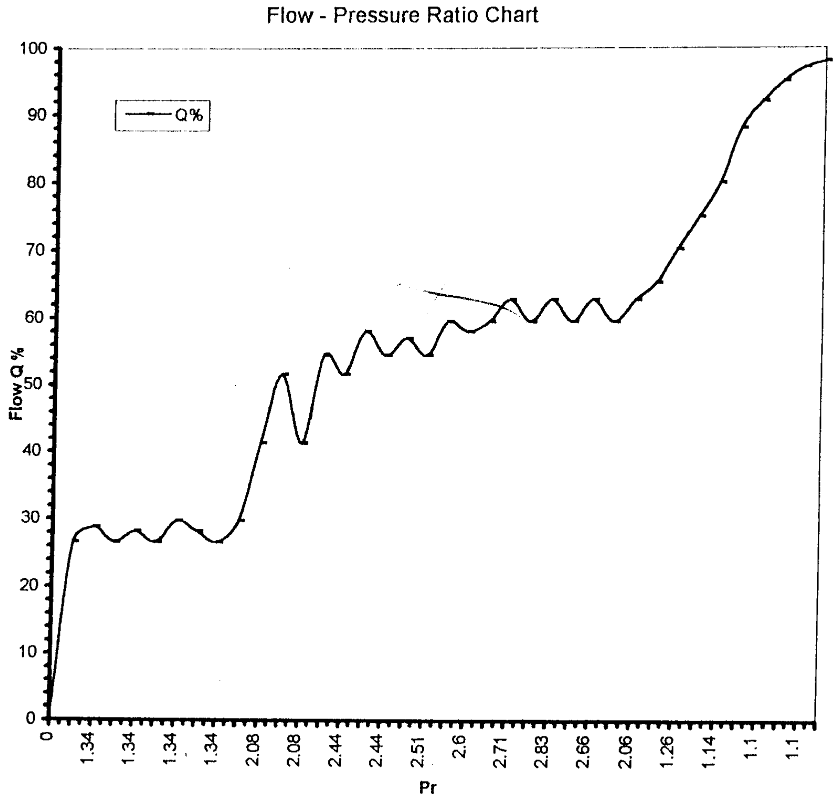


FIG (3-30) EXPERIMENTAL MONITORING OF ROTA COMPRESSION SYSTEM
FOR (FLOW Q% - PRESSURE RATIO Pr) CHART

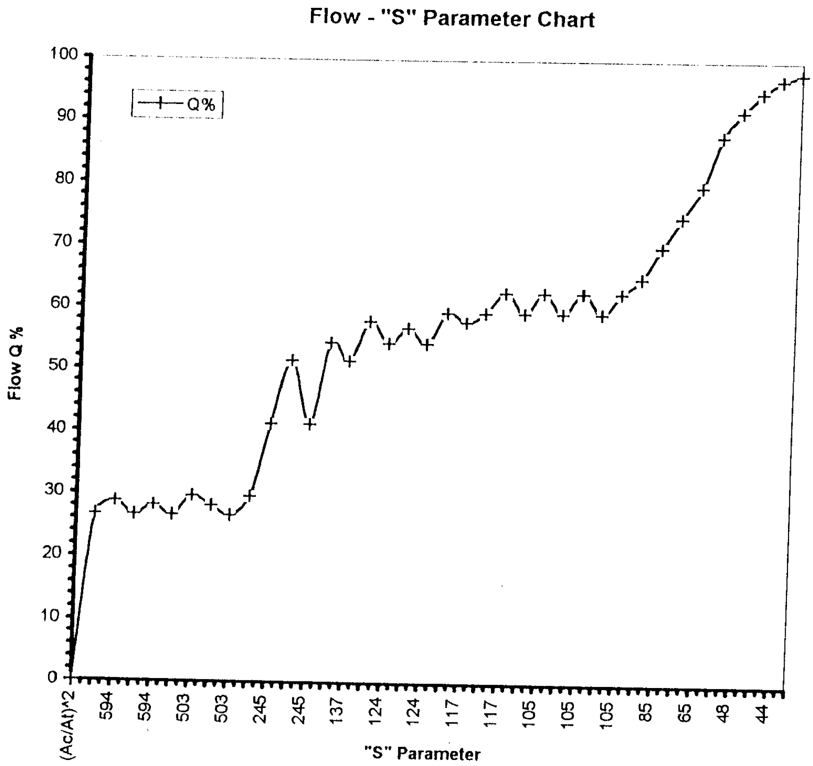


FIG (3-31) EXPERIMENTAL MONITORING OF ROTA COMPRESSION SYSTEM FOR (FLOW Q% -(S) PARAMETER) CHART.

3.5 Anti-*Surge* Control Remarks:

The control systems discussed here could be applied pneumatically or electronically or using computer-based control system, the latter method is the more advanced method used specially in hydrocarbon processing applications the flow chart algorithm is shown in fig (3-34), such system is described simply in fig.(3-35), which illustrates multi input signals as suction pressure, temp., flow and density, and discharge pressure, temp., and motor power and differential pressure (dp) between suction and discharge and (dp) across discharge elbow, also recycle valve position %.

If there is another sharing parallel compressor , the flow signal (h) should be send to the assigned compressor(Anti *surge* controller- XIC).

Effective output signals are to:

1. Suction control valve (JCV) - { Power control valve}
2. Recycle control valve (FCV)- {Flow control valve}
3. Discharge control valve (PCV)- {Pressure control valve}.

This Anti *surge* controller is also called "compressor performance controller" can accomplish the following basic control functions:

1. Executes the selected Anti-*surge* control algorithm.
2. Prevent compressor overload.
3. Control compressor discharge pressure.
4. Minimize control loop interaction.
5. Compressor shut-down and start-up sequences in parallel with another stand alone device called "compressor start-up sequencer".
6. Compute parameters for display like (density, mass flow rate, Mach number, motor

horsepower, recycle valve position, dp, Pressures, temperatures , etc.)

7. Microprocessor used in this case for anti *surge* control system should have scanning time less than 0.1 second at least.

Regarding Anti-*surge* control response time or *surge* avoidance action, most control valves has a stroking time over 45 sec. that is very slow responding time, to improve this slow response time, a volume booster is recommended to be installed between the control valve positioner and diaphragm, this booster will reduce the Stroking time to about 4 sec for faster responding [patlovany, 1986]. In the other hand the recirculated gas must be cooled and passed by *Suction separator* to separate any condensate results from cooling to prevent high inlet temperature from causing head difficulties(Fig 2-13). Anti-*surge* control valve must be large and fast acting, it should be capable of recycling 100% of design flow rate, and the full stroke of the valve should be from 0.5 to 1.5 second, using booster before actuator to increase time response. Anti - *surge* Controller should have Proportional plus integrator (P + I). Derivative control not recommended; it will open the Anti *surge* valve far from the *surge* control line and can cause system oscillations. Rapid oscillations in flow, even in the safe operating zone, can cause the valve to open because the nature of the derivative response. Microprocessors used for Anti- *surge* control must have scanning time less than 0.1 second to enable the DAS logs the *surge* onset (flow oscillations).

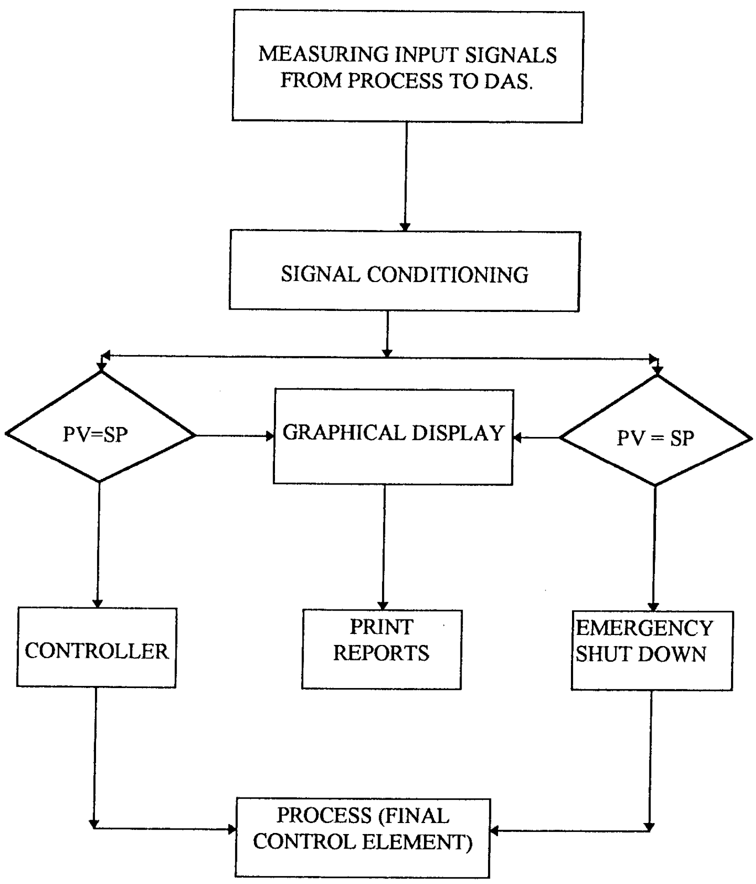


FIG (3-34) FLOW CHART OF PROPOSED (PMC) ALGORITHM.

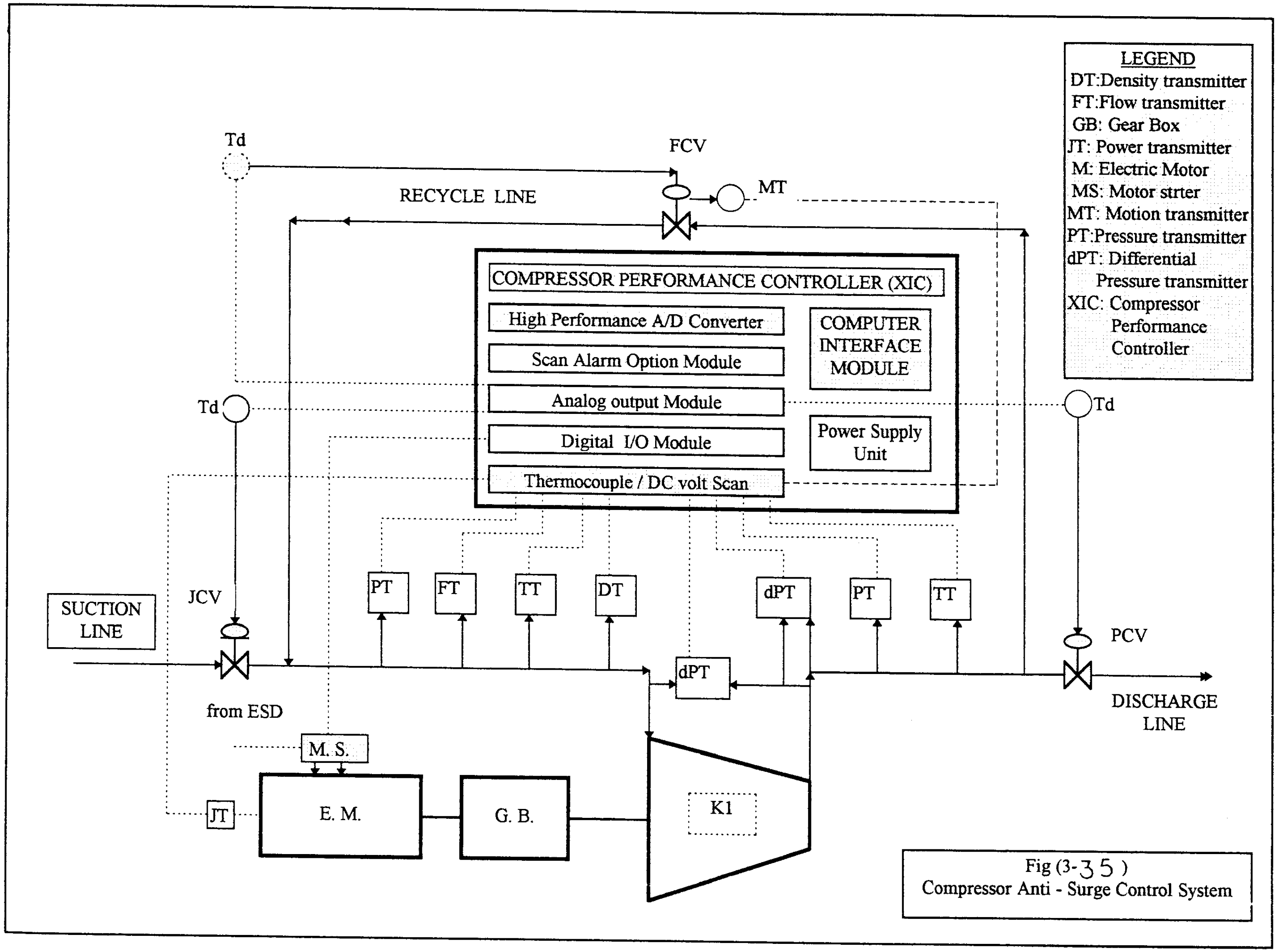


Fig (3-35)
Compressor Anti - Surge Control System

4.1 Discussion:

The analysis of theoretical modeling techniques based on the non linear lumped parameter [Greitzer, 1975, 1981], is useful and powerful specially for axial compressors, during *transient* operation. It is not easy to predict *surge* onset without taking Hansen's parameter (S) in considerations. It considered the main quantitative parameter related to the flow rate through the compression system specially for centrifugal compressors. This analysis answers clearly definite questions; such as for *surge* prediction, when *surge* occurs? Which quantitative parameters are concerning *surge* onset? Which parameters are suitable for Anti - *surge* Control System? Analyzing the *unsteady operation analysis of Rota compression system,* it found that; *Rota* float surging occurs at flow rates below critical value of 63 % of STD reference flow rate and high pressure ratio of Pr = 2.8 at Hansen 's parameter (S > 105) at max. Pressure drop in discharge line downstream the discharge control valve.

Damping occurs after increasing the flow rate above 63 % and (S < 105). The same behavior occurs during throttling the system. comparing these results with Hansen 's results similarity found except the decay of the *Rota* theoretical modeling oscillations. In experimental the oscillation continues at the same rate until the valve (V1) setting changed to the safe limit.

The *Rota surge* occurs at the Greitzer's parameter B > 0.3 that is less than the critical "B" value of 0.6, but Hansen mentioned that the min. value of parameter (B) = 0.105 at low speeds of (7000-9000 rpm), this satisfies this condition under study. There is another reason may cause this compression system to *surge* at value of (B) is less than the critical (B = 0.6). This is due to the obstruction devices (FN, VEN, OP) on the discharge side (up stream of plenum). That makes the (S) parameter is the great concern of *surge* onset, in this case in conjunction of parameter (B).

Discharge flow (Q) and valve parameter (S), are good quantitative parameters for Anti- *surge* control scheme. [Patlovany, 1986] Analyzed a large air compressor for the purpose of minimizing energy costs and protecting the compressor from *surge*. His analysis shows that *surge* occurs for two different multi stage compressors, at

the rate of 66 % and 69.5 % of discharge flow at different rotating speeds (from 4280 rpm to 5110 rpm). This agrees with the results obtained from experiment, that indicate the safe flow rate is more than 63 % for this case. Even the min. time scan of the DAS used was 1 second the turbine meter screen used with the help of stop watch. The visual and audible surging of the *Rota pop* was an enough evidence of *surge* onset instead of using vibration detectors in this case.

Regarding Anti-*surge* control response time, or *surge* avoidance action, most control valves have a stroking time over 45 second. That is very slow responding time. To improve this slow response time, a volume booster recommended to be installed between the control valve positioner & diaphragm. This booster will reduce the stroking time to about 4 sec for faster responding [patlovany, 1986]. In the other hand the recirculated gas must be cooled and passed by *Suction separator* to separate any condensate results from cooling to prevent high inlet temperature from causing head difficulties (Fig. 2-12). Anti - *surge* control valve must be large and fast acting. It should be capable of recycling 100% of design flow rate, and the full stroke of the valve should be from 0.5 to 1.5 second, using booster before actuator to increase time response. Anti - *surge* controller should have proportional plus integral (PI).

Derivative control not recommended; it will open the Anti *surge* valve far from the *surge* control line and can cause system oscillations. Rapid oscillations in flow, even in the safe operating zone, can cause the valve to open because the nature of the derivative response. Microprocessors used for Anti - *surge* control must have scanning time less than 0.1 second to enable the DAS logs the *surge* onset (flow oscillations). The experimental results from this case regarding the safe operating limit of the flow rate (Q = 63%) should be incremented with at least more 5% as a safe margin for steady operation. During experimental monitoring analysis another important parameter appears that the energy losses were max. at *surge* onset (see appendix A), this accomplished with the low flow rates less than 63%, and the peak value of the energy losses is almost equal to the energy gained. This remark may need another separate research.

4.2 Conclusion & Recommendations:

The analytical study of both theoretical and experimental data of the simulated compression system during transient operation was very useful and powerful. This analysis answers clearly definite questions; such as for *surge* prediction, when *surge* occurs? Which quantitative parameters related with *surge* onset? And which parameters are more suitable for Anti *surge* control scheme. Considering the present investigation, the following conclusions and recommendations are concluded:

1. The theoretical modeling done for both Hansen's cases using the modified model, shows typical identity with his published experimental modeling.
2. The theoretical modeling done for *Rota* cases using the modified model, shows reasonable agreement with Hansen's modeling.
3. The theoretical modeling done for *Rota* cases using the modified model, shows a reasonable agreement with the experimental results using hydraulic simulator.

   Greitzer Compressor characteristic's parameter (**B**) has a clear effect on *surge* onset for value of (B = 0.256) that agrees with Hansen's min. Value of (B= 0.105 for low rpm of about 7000 rpm) that is less than Greitzer's value of **B< 0.6** (critical value.)

3. Valve parameter "S" has an effect on *surge* onset for values of **S > 105** (from Experimental logging) This agrees with Hansen's min. value of **S= 47** represents the Stall limit of instability (critical value is S = 125).
4. The presented model in this form is adequate for use in computer based expert system, for its quick responding facility.
5. The using of such model will be of a great help of *surge* prediction and building up the reasonable Anti- *surge* control system.
7. *Rota* float surging occurs at flow rates below critical value of 63 % of STD reference flow rate and high pressure ratio of Pr = 2.8 at valve parameter S > 105 at max. pressure drop in discharge line downstream the discharge control valve.
8. The same oscillations occurs during throttling the system at normal operation, or (experimental data).
9. Comparing the theoretical *Rota* modeling results wits Hansen 's results, similarity found except the decay of *Rota* theoretical modeling charts.

10. The *Rota surge* occurs even the Greitzer's parameter B = 0.256 that is much lower than critical B value of 0.6, this is may be due to obstruction devices installed on the discharge line (up stream of plenum).

11. Damping occurs after increasing the flow rate (experimentally) above 63 % and S <105 (+5%).

12. Discharge flow as a percentage of STD flow and throttling (S) is a good quantitative parameter for Anti- *surge* control scheme.

13. Microprocessors used for Anti-*surge* control must have scanning time less than 0.1 second.

14. It is hopeful to study the change any of the rest of initial geometric data (V_p, A_c, L_c, A_t, L_t, R, N_D, N, L_p) will change the value of (B) and consequently the *surge* onset.

This system used constitute the lab hydraulic circuit control & Simulation that called (simulator-PMC) will provide an efficient and economical mean to simulate real and large industrial process to satisfy the following cases:

- Protect the centrifugal compressor from *surge* due to changing of suction conditions.
- Operate the centrifugal compressor in the safe operating area of the performance curve.
- Maintaining product quality on a continuous bases and with minimum cost.
- Maintain plant production rate at minimum cost.
- Easy maintenance diagnostics that will cut the shut-down periods.
- Assuring the stability of the automated process.
- Avoid human faults allowing to use expert systems.

Finally the obtained results show a reasonable agreement between the theoretical models based on Greitzer and Hansen models and the Simulator experimental results, for prediction and onset of *surge*. But the theoretical behavior analysis of the gas passed through impeller blades shows that the decrease of both absolute radial velocity C_r and angle (α) are far responsible of *surge* onset. That needs to find out new parameter concerning both items noting that C_r is direct function of Flow rate (Q) through compressor, that the main concern here.

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Note :

AIAA : American Institute of Aeronautics and Astronautics.

ملخص البحث

محاكاة تشغيل الآلات الهيدروليكية والتحكم فيها

هذا البحث فى مجال تطبيق أتمتة ومراقبة العمليات الصناعية ومحاكاتها أثناء ظروف التشغيل العادية والإنتقالية ويركز البحث على دراسة الضواغط المركزية الدوارة (التربينية) التى تعتبر من أكبر مستهلكات الطاقة فى مجال الصناعة حيث أنها أنسب وسائل ضغط الهواء والمبردات والغازات سواء فى مصانع البتروكيماويات أو معامل تسييل الغازات البترولية, وقدركز البحث على دراسة ظاهرة الترنح (SURGE) وعمل نمذجة لهذه الظروف الغير مستقره ودراسة أسبابها وطرق علاجها وذلك باختيار النظم الخبيره لمراقبة التشغيل والتحكم وكذلك اقتراح نظام تحكم مستقر لحماية الضواغط من الظروف الغير مستقرة للتشغيل الترنح (SURGE).

وتتكون الرساله من اربعة فصول:

الفصل الأول :(INTRODUCTION)

يمثل مقدمة عامه عن موضوع البحث وأهمية البحث وعرض للأبحاث السابقة فى هذا المجال وأهمية هذه الدراسة والهدف منها وكذلك خطة البحث.

الفصل الثانى:(CENTRIFUGAL COMPRESSORS THEORETICAL ANALYSIS AND MODELING)

ويشمل التحليل النظرى وعمل النمذجة المحاكية لظاهرة الترنح (SURGE) ويتناول تحليل ظاهرة الترنح (SURGE) وذلك بعرض ما يحدث داخل مروحة الضاغط المركزى للغاز ثم يتناول عمل النمذجة والمحاكاة للضواغط المركزية الدوارة أثناء ظروف التشغيل غير المستقرة خاصة ظاهرة الجيشان والتحليل النظرى لها ودراسة آثارها على عمليات تشغيل الضواغط المركزية إعتمادا على نموذج (جريتزر) و(هانسن) بعد تعديله وعمل برنامج فورتران لرسم نموذج (هانسن) عند ظروف تشغيل مختلفة.

ثم يتناول كيفية توقع حدوث الترنح (SURGE) أثناء عدم الإستقرار وعرض طرق التحكم فيها وتجنب حدوثها.

الفصل الثالث :(EXPERIMENTAL WORK)

ويشمل التجارب المعملية ويتناول كيفية تطبيق البحث من خلال وصف عمل جهاز إختبار هيدروليكى وتطبيق عملية الأتمتة والمراقبة من خلال جهاز مسح (جمع) المعلومات الموصل بالكمبيوتر الشخصى.

يتبعها التحليلات أثناء التشغيل المستقر وهى:

أ) المحاكاة باستحدام الكمبيوتر

ب) المراقبه أثناء التشغيل المستقر

ويتبعها التحليلات أثناء التشغيل الغير مستقر وهى:

أ) المحاكاة باستحدام الكمبيوتر

حيث يتم عمل النمذجه بإستخدام برنامج نظام خبير الذى يعتمد على برنامج فورتران يقوم بعمل تكامل من الدرجة الخامسة (بطريقة رنج كوتا RUNG KOTTA) لحل أربعه معادلات (جريتزر) غيرخطيه تصف حاله عدم الإستقرار وإظهار النتائج على منحنيات وتشمل:

١) محاكاة نموذج هانسن (HANSEN).

٢) محاكاة نموذج التجارب (ROTA).

ب) المراقبه أثناء التشغيل غير المستقر

وتشمل رسم المنحنيات الخاصه بنظام الضغط الهيدروليكى.

ويتم تتويج هذا البحث بعرض نظام التحكم فى الترنح (SURGE) كمحصلة لما تم عرضة من تحليلات ونتائج وإقتراح نظام تحكم مناسب وكذلك عمل حزمة برامج مناسبة للمحاكاة والمراقبةوالتحكم.

الفصل الرابع: نتائج البحث والتوصيات (CONCLUSION)

بعد عرض وتحليل النتائج أثناء التشغيل غير المستقر لجهاز الاختبار الهيدروليكى تم التوصل الى النتائج الأتية:

١- تتأرجح عوامه جهاز القياس (ROTA) بشده عندما تنخفض قيمه سريان الهواء الداخل للجهاز عند قيمه أقل من ٦٣٪ من قيمه السريان النمطى، وعند نسبه ضغط تساوى ٢,٨ وعندنسبه فتح الصمام (معامل هانسن) يساوى (S = 125).

٢- يحدث التأرجح سواء أثناء بدايه التشغيل أو أثناء عمل خنق لصمام التصرف.

٣- يستقر السريان عند زياده قيمه سريان الهواء أكبر من ٦٣٪ من قيمه السريان النمطى وعند نسبه فتح الصمام (معامل هانسن) أقل من (S=105).

٤- بمقارنه نتائج النمذجه العمليه مع نتائج النمذجه النظريه

لنظام (هانسن المعدل) المبنى على نظريه (جريتزر) وجدتماثل فى تأرجح التصرف ونسبه الضغط.

وبناء على ماسبق يمكن تقديم التوصيات الأتية:

١ - ظاهرة الجيشان أو الترنح (SURGE) (عدم الإستقرار) فى الضواغط المركزية الدوارة (التربينية) الدوارة تحدث عند إنخفاض مستوى تدفق الغاز من خط السحب وتعتمد على الأبعاد الهندسيه لتصميم الضاغط (معامل جريتزرB) وكذلك نسبه فتح صمام التصرف(معامل هانسن S) ويجب تجنب حدوث الجيشان والترنح (SURGE) وعدم الإستقرار بفتح صمام الراجع بسرعة كافية باستخدام نظام خبير عن طريق كمبيوتر شخصى.

٢ - استخدام نظام المراقبة والتحكم المقترح يتيح الأتى :-

أ - ضمان تشغيل الضواغط المركزية الدوارة (التربينية) بصفة مستديمة فى حالة إستقرار وأمان .

ب - المحافظة على معدل االإنتاج بأقل التكاليف .

ج - المحافظة على معدل نوعية الإنتاج بأقل التكاليف .

د - سهولة التعرف على أسباب مشاكل التشغيل وتقديم أنسب الحلول لفريق الصيانة مما يقلل من فترات التوقف وتوفير نفقاتها .

هـ - تجنب تعرض أفراد التشغيل لمخاطر المراقبة الفردية وتركيز جهودهم بإتاحة إستخدام النظم الخبيرة .

قرار لجنة الحكم

اسم الباحث : مهندس / رسـمى نعمـان محمـد العــزول

عنوان الرسالة: محاكاة تشغيل الألات الهيدروليكيه والتحكم فيهـا.

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م	الأسم	الوظيفة
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تاريخ المناقشة : / / ١٩٩٥م

تقدير الرسالة :

توقيعات لجنة الحكم:

	الأسم	التوقيع
١	الأستاذ الدكتور العدوى شعبان المتولى	
٢	الأستاذ الدكتور مجـدى محمد أبوريـان	
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يعتمد،

عميد الكلية

المشرفون

اسم الباحث : مهندس / رسـمى نعمـان محمـد العــزول

عنوان الرسالة: محاكاة تشغيل الألات الهيدروليكية والتحكم فيهـا.

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بسم الله الرحمن الرحيم

(وعلمك مالم تكن تعلم وكان فضل الله عليك عظيما)

صدق الله العظيم

سورة النساء (آية ١١٣)

جامعة المنصورة
كلية الهندسة
قسم هندسة القوى الميكانيكية

محاكاة تشغيل الآلات الهيدروليكية والتحكم فيها

(كجزء من المتطلبات للحصول على درجة الماجستير فى هندسة القوى الميكانيكيه)

مقدمـــــة من
مهندس / رسمى نعمان محمد العـــزول
بكالوريوس هندسة القوى الميكانيكيه
عـــام ١٩٧٧م

إشـــــــراف

أ . د / مجـــــدى محمد أبوريـــــان
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